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FRICTION
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BY

G. U. WHEELER

WHITWORTH SCHOLAR; ASSOCIATE MEMBER OF THE INSTITUTION OF MECHANICAL ENGINEERS

WITH 62 ILLUSTRATIONS

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PREFACE

THIS small volume is a reprint with re-arrangement and added matter of a series of Articles written for a Technical Journal.

They have been revised and considerable new matter added. The writer had often felt the want of some small and concise *résumé* of the many records of experiments on Friction scattered throughout the proceedings of the various scientific institutions, and also of some outline of the various kinds of lubricants and their properties, with the usual methods of testing them. It is hoped the work will be found useful not only to those actually engaged upon the design of machinery, but also to students and those many users of oils who could not rank themselves as technical men. For the convenience of all, the writer has ventured to depart from the somewhat beaten track in giving a few prices of various oils, and also balls and ball bearings. He has often felt hampered, when wishing to know the comparative cost of various methods, to find all such information sedulously excluded from Engineering literature.

It only remains to express the writer's indebtedness to such standard works as those of Allen, Wallace, and Veitch Wilson, in reference to the Chemistry and Testing of Oils, and to Thurston and the Proceedings of the Institution of Mechanical Engineers for actual experimental work.

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FRICTION AND ITS REDUCTION

CHAPTER I

INTRODUCTORY

Introductory—Heads of the Subject—Conditions to Consider in Selecting Lubricants—How Losses Occur—Lubricating Methods: Results of Carelessness—Waste of Oil and its Results—Clogging—Hints on Purchasing Lubricants—Investigations as to the Losses through Friction, etc.—How to Treat the Subject.

IT has been remarked that "the study of the methods and magnitudes of friction-losses and the laws governing their production is, next to the theory of pure mechanism, the most important study in relation to the transmission of power by machinery." Loss of power means, of course, loss to the pocket, larger coal bills, and larger expenses in various ways; and in showing how to save, it is hoped that some hints will be given which will prove of substantial benefit. The writer will endeavour, therefore, to point out how these friction losses may be reduced to a minimum, and will deal with the various mechanical devices for reducing friction, as well as with the various frictional agents comprised within the nomenclature of oils and lubricants.

When considering how to reduce friction, we should lay down the rule that it is not simply the initial outlay which is to be taken into account, but the ultimate gain or loss in expenditure resulting from the means adopted. Again, it cannot be too strongly enforced that, in this matter, careful attention to small—and to what many would consider insignificant—details is productive of substantial and continuous economy.

Heads of the Subject.—This subject naturally divides itself under two great heads, namely, (1) the application of lubricants whether liquid or solid; and (2) the introduction of mechanical devices in the rubbing parts themselves, such as friction bearings—where the revolving parts run upon rollers or balls—or by the use of anti-friction metals. The use of ball bearings, it may at once be said, will undoubtedly very rapidly increase now that both in England and across the Atlantic machinery has been perfected to turn out hard steel balls, which answer perfectly all the severe wear to which they are subjected. Seeing, moreover, that an American firm has successfully fitted up mine trolleys with this class of bearing, there seems no reason why a similar application cannot be made to very many other purposes from which the excessive cost has hitherto excluded them.

Conditions to Consider in Selecting Lubricants.—In connection with the use of lubricants, due regard must be had to *the class of work* they have to perform; *the situation* in which they are used; *the weight of the machinery* they are lubricating; *the temperatures* to which they may be subjected—for it is evident that if machinery is out in the open and exposed to cold, care must be taken that the lubricant maintains its fluidity at low temperatures. Again, the oil that would be very suitable for a loom would never do for the bearings of a locomotive, and an oil that would be effective for the bearings of a lathe would perhaps never stand the excessive temperature to which it would be subjected if used in a steam-engine cylinder supplied with steam of 100 lbs. pressure.

How Losses Occur.—In considering how to reduce losses due to friction, three things have to be borne in mind: (1) Expenditure for oil; (2) expenditure for coal; (3) expenditure for repairs. These three items run very closely together, and the last two may be very materially affected by the first. To show how the items of oil and coal are related, we may assume, for example, that an oil of too great body or "viscosity," as it is technically termed, has been chosen; then extra work is thrown upon the engine, and more steam and consequently more coal is required to overcome the drag put upon bearings, etc., owing to the lubricant no longer reducing the friction, but acting as a brake upon the machinery. Again, suppose an oil has been chosen which is liable after a time to decompose or become rancid, thus developing acids which will attack metals, the result will be an increase of expenditure for repairs. Or, again, if an oil is used which contains any mechanical impurities (and many dark oils so generally used for cylinders do contain such, their very darkness making it difficult to detect them)—then it is obvious that abnormal wear will take place, causing outlay for repairs which might have been easily avoided. An instance is before us where, in the case of an engine of 1,000 indicated horse-power—a large power it may be said, but it will forcibly illustrate the point—an increase of 2½ tons of coal per day of ten hours was necessitated by the use of a poor oil as compared with a good one. Another case, where an oil used was found to contain 5·5 per cent. of free acid, produced in the cylinder of a steam engine, after six months' use, a thick matter which, when analyzed, yielded 6 per cent. of metallic iron, being the measure, as the investigator said, of the destructive friction between piston and cylinder during that period.

Lubricating Methods: Results of Carelessness.—The manner of the introduction of lubricants into cylinders, bearings, etc., calls for much careful attention, and has much to do—we might almost say everything to do—with the success which may accompany the use of any particular oil. A dry place occurring on a journal will instantly cause heating and abrasion, necessitating, perhaps, considerable delay and expense in its rectification. The delay in these cases often causes more trouble and annoyance than the mere item of expense for repair. A few years ago a case occurred in the writer's experience which will illustrate this. A crane had a long slow-moving bearing, difficult of access; this had been thought of in the drawing-office, but in the shops, the provision made for lubrication was neglected, and, as so often happens, the first intimation received of the omission was that the bearing had worn out of truth over half-an-inch in a few weeks, thrown a pinion out of gear, and stripped the teeth off a rack. This was entirely due to neglect in the distribution of the lubricant to the bearing.

Waste of Oil and its Results.—There is much carelessness on the part of those in charge of engines and machinery in the amount of oil they introduce, and in the case of condensing engines, where the feed is taken from the hot-well, much damage is often done to boilers by excessive deposits of grease on the furnace tops. Where condensed water is used for feed, a certain amount of oil must pass with the water into the boiler, in spite of all the precautions that can possibly be taken to strain out the oil by mechanical means—such as filters—so that if more than is actually necessary for efficient lubrication be permitted to pass into the cylinders, a corresponding greater amount will pass into the boiler. An instance has recently come under our notice where the rivets at the top of the furnace mouths of a "Lancashire" boiler were continually leaking, and even after larger ones were put in with counter-sinks the same thing occurred. This was found to be caused by the excessive deposit upon the top of the furnaces of a greasy nature, and such deposits are very impervious to heat, so that they got overheated, causing the top of the furnaces to expand to an abnormal extent, and straining the front plate of the boiler at the juncture of the furnaces, thus loosening the rivets.

Clogging.—It will be familiar to every one how a sewing machine will become clogged up after standing unused for a while owing to the thickening of the oil. We refer to this only to point out a class of machinery where the power used to perform the work done by the machine is not great as comparable to the frictional resistances of the machine under bad or insufficient lubrication. The "gumming" of the lubricant is due to oxidation—that is, to the absorption of oxygen from the atmosphere, and is taken advantage of in the case of linseed oil for paints; but, of course, for lubrication an oil which shows any tendency to gum is inadmissible. Accompanying oxidation there is always the generation of heat, and every one is familiar with the way in which a mass of waste will heat and fire when saturated with a certain class of oil—and this spontaneous combustion is often fraught with most terrible consequences if not discovered in time. Many large fires which have been unaccounted for are most likely to have been caused by some oily waste thrown inadvertently near a steam pipe, the heat from which hastened the combustion. But though animal and vegetable oils are very liable in this respect, yet mineral oils show no tendency whatever to do so, and if added to the extent of 50 per cent. to animal or vegetable oils, will materially reduce the tendency of the oil to ignite spontaneously, in some cases entirely preventing it.

Hints on Purchasing Lubricants.—From the foregoing general survey it will be seen that there are many things to be taken into consideration when purchasing lubricants and lubricating appliances, that, at first sight, do not appear to connect themselves with the matter. Often an oil offered at a cheap rate will in the end prove to be a very costly one—dear if it could be had for nothing. In every business an article to sell cheaply has to be produced cheaply, and processes of manufacture have to be omitted which are essential to the production of the *best.* So with the manufacture of lubricants, they can be produced for a low figure, and even when they are required to meet a specification, there is such a divergence in the price as can be accounted for only by supposing that some makers require an inordinate profit, or, what is more likely, that some supply an inferior article doctored up to meet the specification tests.

Investigations as to Losses through Friction, etc.—There has been no superabundance of investigators in the field of the actual losses incurred by various ways in machinery; but both in England and America, of late years, this subject has been taken up, and the results will be given in proper course. As early as the year 1830, General (then Captain) Morin carried out an elaborate series of experiments to determine the laws governing friction; but in these the materials were the chief thing under consideration, whereas with machinery the chief consideration is, knowing what will be used (and they are very few), what will be the best and most suitable lubricant under varying conditions of load and velocity. It is in this respect that we speak of there having been but few investigators in the field. Hirn in 1855 made experiments upon shaft bearings. There have been many instruments devised and perfected for testing the viscosity, flashing point, co-efficient of fluid friction, and other properties of oils, which will be briefly described in their place.

How to Treat the Subject.—As an indication of the manner in which it is proposed to treat the subject, we lay the following before readers' attention as a general syllabus:—

I. *Friction*, as far as it concerns Machinery; Laws governing Friction; Measurement of Friction; brief account of various Experimenters on Friction.

II. *Methods adopted for the Reduction of Friction.*—(*a*) Lubricants; (*b*) Mechanical Devices.

Under (*a*) will be taken up the various kinds of materials used as lubricants, viz. water, oils, and graphite. Of these, oils will naturally be treated at far greater length than the other two, these being seldom used. Some, indeed, might not consider water as a lubricant, yet when shafts run in lignum vitæ bearings, as in the stern tubes of screw-ships, water is entirely depended upon to prevent heating or abrasion at that particular part.

Animal, vegetable, and mineral oils, both alone and as compounds, will be considered, and the tests used to determine decomposition ; ignition and vaporization ; temperatures, co-efficient of friction, acidity, and such other properties as may be necessary to be known by users.

Since what are commonly known as " solid " oils are coming to the front, and great economy claimed for them under certain conditions of work, they will receive our attention, practical examples of their use, extending over considerable lengths of time, being adduced.

After this will be considered the best oils to be used under given conditions, such as for steam and gas engine cylinders, electric light machinery, slow and fast running machinery, etc., etc. ; use, again, of oil that has been caught from bearings, etc.

This section will close with what is, as before stated, a most important factor, viz. the distribution of lubricants, and the regulation of their supply ; so that under various conditions *proper* lubrication is, as far as possible, automatically supplied. A writer of authority upon this subject has said that shafts running in self-adjusting cast-iron bearings, the bearings being four diameters long, at a speed of 120 revolutions per minute, require on an average 2·2 fluid oz. of oil per bearing every six months. This is a very small quantity when compared with the plentiful amount generally bestowed upon shafting bearings.

Under the head (*b*) will come most prominently roller and ball bearings, and then anti-friction metals, as they are called, being various compositions which reduce the friction between the moving parts by replacing one of them by a softer metal, which is held in a casing of harder material, or put in the bearings in narrow strips.

CHAPTER II

FRICTION

Friction—Kinds of Friction—Solid Friction (Sliding)—Illustration—Laws of Sliding Friction—Solid Friction (Rolling)—Laws of Rolling Friction—Fluid Friction—Laws of Fluid Friction.

FRICTION is the name given to the well-known resistance to motion which is manifested when two bodies in contact are moved relatively to one another. It is due to the interlocking of projecting particles upon the surfaces in contact, and the power expended in overcoming friction is entirely due to the fact of having constantly, as it were, to unlock these projecting particles or asperities. When, however, instead of unlocking, they adhere firmly one to the other, then, for motion to take place, one set of particles has to be torn off from one or other of the surfaces, producing what is usually known by the term *abrasion*. In theoretical mechanics perfectly smooth bodies, or bodies with perfectly smooth surfaces, are sometimes assumed, but this is not realized in the actualities of this lower world of fact. However smoothly surfaces may be got up, there is always a certain resistance to motion displayed, which resistance it is our business, by lubrication or otherwise, to reduce to the lowest point possible.

Kinds of Friction.—There are two kinds or varieties of friction, very clearly defined and governed by two totally distinct sets of laws:—

I. Solid Friction.
II. Fluid Friction.

Solid friction is that which is found to act between dry solid bodies, or, if not quite dry, with very slightly lubricated surfaces. It is generally divided into two parts:—

(*a*) Sliding or Gliding Friction; and
(*b*) Rolling Friction.

Fluid friction is that resistance to motion exhibited when liquids—and gases too, for they come under the head of fluids—are caused to move in themselves or upon solids.

Compound Friction.—These two varieties of friction are often combined one with another, as when, by the introduction of lubricating fluids between surfaces in contact, there is not only the friction from sliding simply, but also that from the fluid to take into account, and, as these may both vary, the laws governing such compound friction are necessarily far more difficult to decide and enunciate than either those of solid or fluid friction alone. Under this head come most of the frictional resistances experienced in machinery, such as well-lubricated bearings, etc.; but before proceeding further with it, we will briefly consider the two simpler cases.

Solid Friction.—(*a*) *Sliding Friction.*—The first thing to be understood is the method by which sliding or gliding friction is measured. This is done by reckoning the relation which the force just sufficient to produce motion bears to the weight of the body set in motion.

Thus in Fig. 1, if a body of weight W lbs. rests on the surface A B, and a weight of P lbs. is found necessary just to produce motion, then the relation

$$\frac{P}{W}$$

is called the co-efficient of friction, and is generally denoted by the letter f. Thus, if W be 1 lb., and the surfaces in contact be metals and dry, P has been found by experiment equal to 0·2 lb., after allowing for the friction of the pulley and cord. And if W is increased, P will increase in the same ratio; so that, knowing the co-efficient of friction in any one case as above, the amount of friction, or the force necessary to overcome the friction, will be found by multiplying the weight by 0·2. It must be borne in mind that this co-efficient applies only in the case of metal on metal, the surfaces being dry and smooth. Full tables of co-efficients will be given further on.

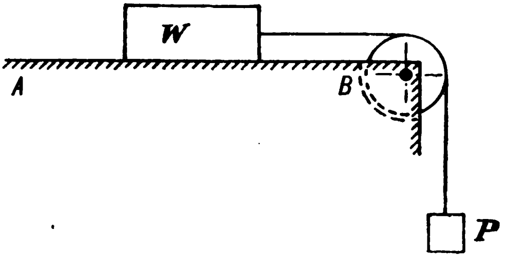


FIG. 1.

Another way of determining the co-efficient of friction is by the use of an inclined plane, as shown in Fig. 2, so arranged that the angle which it makes with the horizontal can be adjusted until the substance upon it will no longer remain at rest; then the relation

$$\frac{C\,B}{A\,B}$$

(C B being perpendicular to A B) will be found to be exactly equal to

$$\frac{P}{W},$$

as in the previous method.

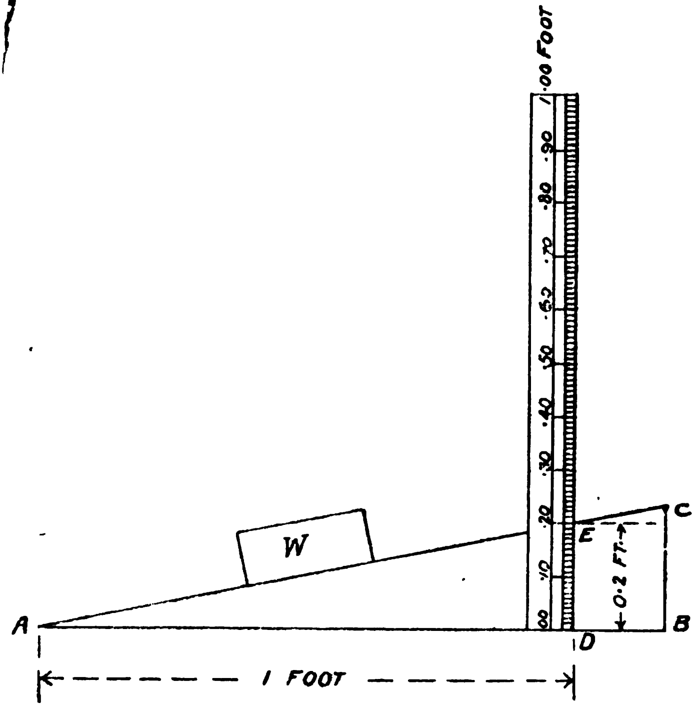


FIG. 2.

To make it very simple, let A D be exactly one foot long, and at D let a scale one foot long, divided into 100 parts, be placed as shown ; then, if the surfaces are metals and dry, it will be found that the body will begin to slide when the vertical distance D E is about twenty-hundredths. That is, the co-efficient of friction is equal to

$$\frac{20}{100} = 0{\cdot}2 = f.$$

Having now found a way of measuring the resistance due to friction, it is very easy to calculate the amount of work absorbed in overcoming it. It is most convenient for practical purposes to express this in horsepower.

$$\text{Horse-power} = \frac{\left.\begin{array}{l}\text{Resistance to be}\\ \text{overcome in lbs.}\end{array}\right\} \times \left\{\begin{array}{c}\text{Velocity in ft. per minute}\\ \text{of moving body.}\end{array}\right.}{33{,}000.}$$

Illustration.—For an illustration let us take an ordinary slide valve, as shown in Fig. 3.

Here the pressure exerted by the steam upon the face of the valve is area of valve in square inches multiplied by pressure in lbs. per square inch.

$$12\tfrac{1}{4}'' \times 8\tfrac{1}{4}'' \times 100 = 10{,}106 \text{ lbs.}$$

The valve weighs, say, 50 lbs.

From the sum of these must be deducted the upward pressure exerted upon the valve by the steam in the exhaust port, at, for convenience, say, 40 lbs. pressure. This is

$$10'' \times 4'' \times 40 = 1600 \text{ lbs.}$$

$$\begin{array}{r} +10106 \\ +50 \\ -\underline{1600} \\ 8556 \text{ lbs.} \end{array}$$

Multiply this by 0·2, the co-efficient of friction generally taken, although, if the valve be well lubricated, it may be less. This gives us the resistance to be overcome in lbs.

$$8556 \times 0{\cdot}2 = 1711{\cdot}2 \text{ lbs.}$$

If the revolutions of the engine be 100 per minute, and the travel of the valve 3″, the average speed of the valve in feet per minute is

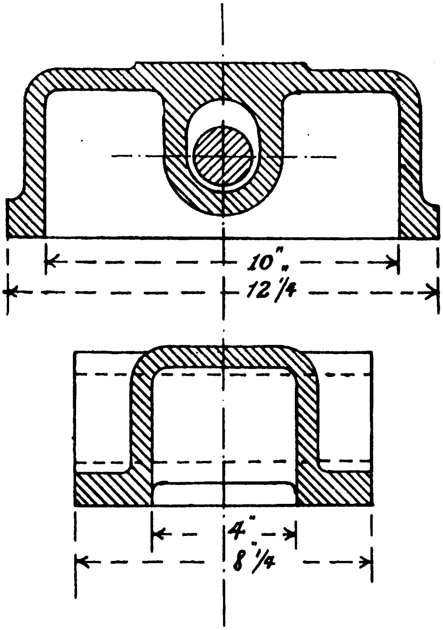


Fig. 3.

$$\frac{100 \times 3 \times 2}{12} = \frac{600}{12} = 50 \text{ ft.}$$

Now, placing these values in the formula above, we have

$$\text{Horse-power} = \frac{1711{\cdot}2 \times 50}{33{,}000} = \frac{85{,}560}{33{,}000} = 2{\cdot}6.$$

This may seem an abnormal power, but when the slide valve is very dry it will not be far out. If the valve could be kept well lubricated, a smaller co-efficient of friction than 0·2 could be used.

Laws of Sliding Friction.—There are generally given in text-books on Mechanics the three so-called "Laws of Friction." They are—

I. The amount of friction is dependent upon the nature of the materials in contact, and, when these remain the same, is in direct proportion to the pressure between them.

II. The amount of friction is independent of the extent of the surfaces in contact so long as the "total" pressure between them remains the same.

III. The amount of friction is independent of the velocity when the body is in motion.

To which may be added also—

IV. The friction of rest is greater than that of motion.

These "laws" are due to Coulomb, who gave publicity to them in 1785, and they often bear his name; but a name far more familiarly associated with them is that of the late General Morin, who, in the years 1830–1834, made at Metz a long and elaborate series of experiments with nearly every variety of constructive material. The apparatus which he used was in principle the same as that shown in Fig. 1, and his results verified the truth of Coulomb's laws. But it is important in all such experiments always to bear in mind the limits between which they were made, and not to consider that universal, which has only been proved within comparatively narrow limits. The limits of velocity and pressure between which Morin conducted periments are exceeded now in every-day practice; instance, the greatest pressure in Morin's case veen the surfaces was scarcely 130 lbs. per square , and now-a-days 800 to 1,000 lbs. per square h are not considered excessive pressures for some arings.

During the last twenty years, however, there has een considerable attention directed to the subject by English and American engineers, and the light thrown upon it by their investigations has shown, beyond doubt, that these "three laws" are only true within the limits between which the experiments were conducted; and, strange to say, that almost immediately Morin's limits of pressure and velocity are exceeded, divergence from his results becomes unmistakably manifest. Again, the temperature does not seem to have been taken into account in his experiments, but M. Hirn, in 1855, showed that the co-efficient of friction diminishes when the temperature is raised.

The results of these more modern experiments with regard to friction will be more in place when we come to make a rapid survey of the work of Prof. Thurston, Messrs. Galton and Westinghouse, Mr. Woodbury, Mr. Beauchamp Tower, Mr. Stroudley, Mr. Goodman, and others. The greatest honour must of course be given to the one who first conducted any reliable experiments, and this, we are assured, will be given readily to the late General Morin by all those who have followed in his footsteps, however much more light they may have contributed to the points he sought to elucidate.

In a very interesting letter, dated March 15, 1879, addressed to the Secretary of the Institution of Mechanical Engineers, General Morin writes:—

"The results furnished by my experiments as t relations between pressure, surface, and speed, on one hand, and sliding friction on the other, have al been regarded by myself not as mathematical laws, as close approximations to the truth, within the li of the data of the experiments themselves. The sa holds, in my opinion, for many other laws of practi mechanics: such as those of rolling resistance, flu resistance," etc.

Solid Friction.—(*b*) *Rolling Friction.*—Under this head comes the resistance experienced in causing one body of circular section to roll upon a surface so that successive points on the rolling body come into contact with the other surface. Remembering what was said as to sliding friction, that the asperities, or points on the surfaces, had continually to be unlocked or lifted over one another, it will clearly be seen that in rolling friction they come into contact more as a wheel with teeth upon it would if rolled along a rack. Thus the friction is considerably less than in the case of sliding friction. The well-known use of rollers under a log shows that the principle was acted upon from time immemorial, if not stated in formal and explicit language. There is also, in rolling friction, to be taken into account the work absorbed in producing the rut that is so well illustrated when a cart-wheel runs on a road; and even should the material return, in virtue of its elasticity, to its original surface, yet there is this resistance to be overcome. When the surfaces are very hard this is not so appreciable, but if one or both of the surfaces are very yielding, it very materially affects the co-efficients of this kind of friction.

Laws of Rolling Friction.—The following is an approximate statement of the laws governing rolling friction:—

I. The amount of friction is considerably smaller than sliding friction.

II. The amount of friction is proportional to the pressure.

III. The amount of friction is inversely proportional to the diameter of the rolling body (as may easily be seen from the fact that a cart with large wheels is much easier to draw along then the same one with smaller wheels).

When we speak of wheels or rollers for reducing friction, we have to bear in mind that they very nearly always run upon axles in bearings. This of course brings into consideration the sliding friction of the bearings, and therefore—

IV. The amount of friction when a roller runs in bearings is less the smaller the radius of axle, and greater the radius of the wheel.

Fluid Friction.—Strictly speaking, there is no friction among the particles composing liquids or gases as we generally understand friction; this is evidenced by the fact that variation of pressure on the liquid or gas does not affect the friction observed, whereas when solids are experimented upon, it is a matter of every-day observation that (everything else remaining the same) when the pressure brought to bear upon two bodies in contact is increased, the amount of friction experienced in sliding one upon the other is increased also. This leads us to the conclusion, that in the case of fluids the particles can move upon one another without friction. It is, therefore, generally considered among scientific men that fluid friction is caused by the eddies or whirls which are set up in the liquid, such eddies absorbing energy from the body causing motion.

Laws of Fluid Friction.—Professor Thurston, who has made a great variety of experiments as to it, gives the following laws as governing this kind of resistance to motion:—

I. Fluid friction is independent of the pressure between the masses in contact.

II. The resistance of fluids is directly proportional to the area of the surface exhibiting it.

III. This resistance is proportional to the square of the relative velocity at moderate and high speeds, and to the velocity nearly at very low speeds.

IV. It is independent of the nature of the surfaces of the solid against which the stream may flow; but it is dependent to some extent upon the degree of roughness of those surfaces.

V. It is proportional to the destiny of the fluid, and is related in some way to its viscosity.

CHAPTER III

FRICTION (*continued*)

Experiments of Galton and Westinghouse, Prof. Fleeming Jenkin, and Mr. Beauchamp Tower—Compound Friction—Experiments with Oil Bath.

Experiments of Messrs. Galton and Westinghouse.—In the year 1878 Captain Douglas Galton read a paper before Section G of the British Association giving the results of some experiments carried out by him in conjunction with Mr. Westinghouse upon the frictional resistances of brakes on railway trains. Through the courtesy of the London, Brighton and South Coast Railway Company, and the co-operation of their locomotive superintendent, the late Mr. Stroudley, M.I.C.E., these experiments were conducted on the line belonging to the above company. A carriage or van was also made specially for the purpose, having all the necessary recording instruments. Three sets of experiments were made: (*a*) between cast-iron brake-blocks and the steel tyres of the wheels; (*b*) between wrought-iron brake-blocks and steel tyres of wheels; and (*c*) the wheels being skidded by the brakes between the steel tyres and the steel rails. It is almost unnecessary to remark that the surfaces were dry, and that deductions made from the following data apply to that division of our subject.

The following is a table furnished by Captain Galton giving the observed co-efficients of friction between cast-iron brake-blocks and the steel tyres of the wheels at different velocities:—

TABLE I.

Average.		Co-efficient of Friction between C. I. Brake-Blocks and Steel Tyres of Wheels.			
Miles per hour.	Feet per second.	From commencement to end of first 3 seconds.	At from 5 to 7 seconds.	At from 12 to 16 seconds.	At from 24 to 25 seconds.
Under 5	7	·360	—	—	—
„ 10	14	·320	·209	—	—
„ 20	29	·205	·175	·128	·070
„ 30	43	·184	·111	·098	—
„ 40	58	·134	·100	·080	—
„ 45	65	·125	—	—	—
„ 50	73	·100	·070	·056	—
„ 60	88	·062	·054	·048	·043

From the above data we gather two important facts as to the laws governing friction. *First,* that the co-efficient of friction between the surfaces in motion is reduced as their relative velocity, *i. e.* their velocity with respect to each other, is increased. *Second,* that the co-efficient of friction at any one speed, say 20 miles an hour for example, becomes gradually less as the time the brake was in contact with the wheel was lengthened. Thus the co-efficient observed during the first three seconds, in the case when the velocity was 20 miles an hour, was ·205. But between the 5th and 7th seconds of contact was only ·175. Again, between the 12th and 16th seconds it was reduced to ·128, and again to ·07 between the 24th and 25th seconds of contact. Captain Galton remarked that "The longer the time the surfaces are in contact, the smaller apparently does the coefficient of friction become. This result appears more marked in the case of cast-iron blocks than of the wheel sliding on the rail; at all events for the first 30 seconds of the contact, the arrangement not admitting of the experiments being carried on for a longer time. This effect, however, does not appear to be unnatural, as the friction develops heat, and the consequent expansion tends to close up the pores, and to make the heated surface a more united surface than the colder surface. Besides which, it is probable that, in the act of rubbing, small particles may be detached which may act as rollers between the surfaces."

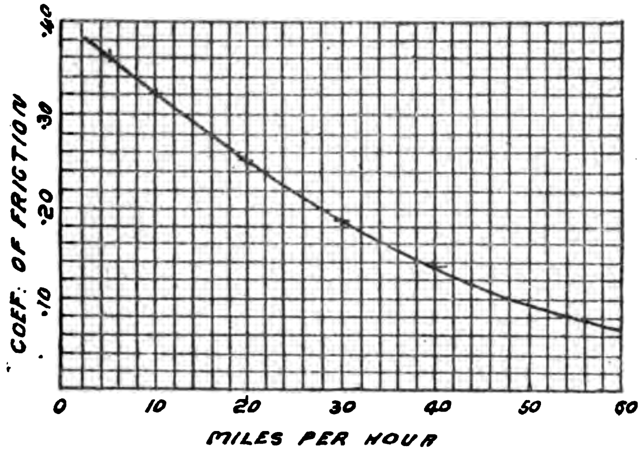


FIG. 4.

The above diagram (Fig. 4) shows a curve plotted out by the author from the values given in Table I., the perpendicular distance from the base line to the curve giving the value of the co-efficient for that velocity. It will be seen that the decrease in the value of the co-efficient is less marked as the velocity increases.

There are given also two more tables, corresponding to the last two sets of experiments (*b*) and (*c*), of which those below are abridgments.

TABLE II.

Average.		Co-efficient of Friction between wrought-iron Blocks and Wheels.
Miles per hour	Feet per second.	From commencement to end of first 3 seconds.
18	26	·170
31	45	·129
48	70	·110

TABLE III.

Average.		Co-efficient of Friction between Steel Tyres and Steel Rails when wheels were skidded.
Miles per hour.	Feet per second.	From commencement to end of first 3 seconds.
10	14	·110
15	21	·087
25	36	·080
38	55	·057
45	65	·051
50	73	·040

Although we have not reproduced the figures in the above tables (II., III.), yet in Captain Galton's the same decrease in the co-efficient is noticeable as the time the brakes were in contact was lengthened. We have already given his judgment as to the cause.

Experiments of Prof. Fleeming Jenkin.—In the Proceedings of the Royal Society for March 22, 1877, there is an abstract from a paper contributed by Prof. Fleeming Jenkin on "Friction between Surfaces moving at Low Speeds." The results given, if not exactly important to us, treating the subject as we are from a commercial standpoint, are yet so exceedingly interesting that we briefly bring them under our readers' notice.

They occupy a position as far on the one side of Morin's experiments as those just described do on the other, Captain Galton's velocity being very much higher than Morin's, and Prof. Jenkin's being at velocities so low that Morin's apparatus could not deal with them. It had occurred to Prof. Jenkin that there might very possibly be some kind of continuity between the friction of rest and the friction of motion, and that there was not that abrupt change in the co-efficient of friction, when motion began, which had generally been assumed. His idea was that, as the relative motion of the surfaces became gradually slower, the co-efficient would increase also until it reached the larger value hitherto known as the co-efficient of rest.

The limits of velocity between which the determination of the value of the co-efficient was made were 0·01 foot per second down to 0·0002 foot per second. Some idea of the low velocity may be gained if we point out that, assuming a body continued to move with that velocity for a period of 1 hour 23·3 minutes, it would have just gone *one foot*. At the higher velocity the body would cover one foot in 1·3 minutes. The surfaces moving upon one another were—

Steel on steel	Under all of the three conditions	Surfaces dry.
Steel on brass		„ oiled.
Steel on agate		„ wet with water.
Steel on beech		
Steel on greenheart		

He thus concludes:—"In the cases steel on beech, oiled or wet with water, and steel on greenheart, oiled or wet with water, the co-efficient of friction increased as the velocity diminished between the two limits given above, the increase amounting to about 20 per cent. of the lower value. It appears that at the higher limit of velocity there was little further tendency to change in the co-efficient, but it is impossible to say how much additional change might have taken place between the lowest limit of velocity and rest. In the case of steel on agate wet with water, there was a similar but much less marked increase as the velocity decreased. In all other cases the friction seemed to be perfectly constant and independent of the velocity. Out of all the sets of circumstances investigated, the only ones in which there was a large difference between the static and kinetic (rest and motion) values of the co-efficient of friction were those in which a decided increase was observed in the kinetic value (that of motion) as the speed decreased. This result renders it exceedingly probable that there is continuity between the two kinds of friction."

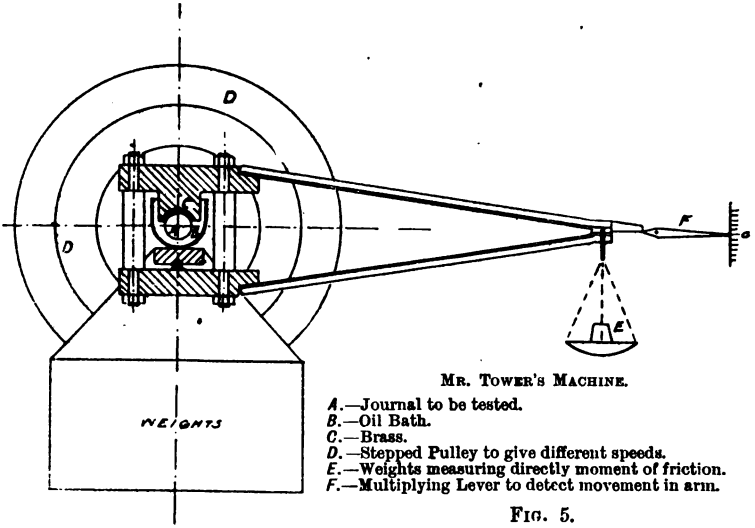
The following table will present more graphically to the eye the relation in which these three sets of experiments stand to one another than any amount of descriptive matter:—

Name of Experimenter.	Lowest Velocity.	Highest Velocity.
Jenkin . . .	·0002 ft. per second . .	·01 ft. per second.
Morin . . .	1 ft. per second (about) .	10 ft. per second.
Galton . . .	7 ft. per second (about) .	88 ft. per second.

Compound Friction.—The experiments just adduced were in connection with solid friction, but the account of the experiments that follows deal with compound friction, as it is exhibited in the case of lubricated journals. It will be recollected that we did not enlarge on compound friction when it was referred to on

p. 11 because, to properly treat of it, the laws governing the two varieties of friction, solid and fluid, must be, at least, briefly treated of first. It has been sufficiently clearly explained that compound friction is a combination of solid and fluid friction, and now our readers will readily follow us when we say, that as the conditions of lubrication more nearly approximate to either of these, so the results will show that the frictional resistances are governed by the laws which are observed to act in that kind of friction to which it approximates. The method, then, by which we intend to point out the laws governing compound friction is to briefly summarize the results of the most important investigators. For laws, after all said and done, are only abstractions from sets of uniform facts, so that given the facts we are better able to judge what would occur under slight modifications of conditions than if we were tried by rigid and inflexible rules.

Mr. Beauchamp Tower's Experiments.—In the year 1883 Mr. Beauchamp Tower presented to the Institution of Mechanical Engineers a report upon certain experiments on "The Friction of Lubricated Bearings and on the Value of Different Lubricants," made by him under the auspices of that institution. The journal experimented upon was 4 in. diameter by 6 in. long. The general arrangement of the machine which was used is shown in Fig. 5. Although experiments were made as to the relative value of different methods of applying the lubricant, yet it was deemed necessary to have a standing method that might be easily and accurately reproduced again and be at the same time uniform in its action. This was found to be fulfilled by the oil bath, that is a receptacle under the bearing containing oil, so that the under side of the shaft is always kept in contact with the oil. It will be seen in the figure that the brass experimented with embraced only one-half of the journal, leaving room for the oil bath underneath. It is impossible, so far as we see, to obtain a more perfect method of lubrication than the oil bath, so that the results obtained from this method of lubrication may be considered the best attainable, every condition being taken that possibly could be to reduce the value of the co-efficient of friction.



Mr. Tower's Machine.

A.—Journal to be tested.
B.—Oil Bath.
C.—Brass.
D.—Stepped Pulley to give different speeds.
E.—Weights measuring directly moment of friction.
F.—Multiplying Lever to detect movement in arm.

Fig. 5.

Experiments with Oil Bath.—*The first point to be noted* in the oil-bath experiments is that the co-efficient of friction became less as the load per square inch on the brass became greater, the velocity of the bearing and its temperature remaining the same. Within ordinary working limits the co-efficient varies inversely as the load; that is, if the load be doubled, the co-efficient will be halved. We have picked out two cases to illustrate our meaning from among those given by Mr. Tower.

CASE I.	Bath of olive oil.	Revs. per min., 200,
	Load per sq. in., 258 lbs.	Co-eff. of fric., ·002,
	" " 520 "	" " ·001.

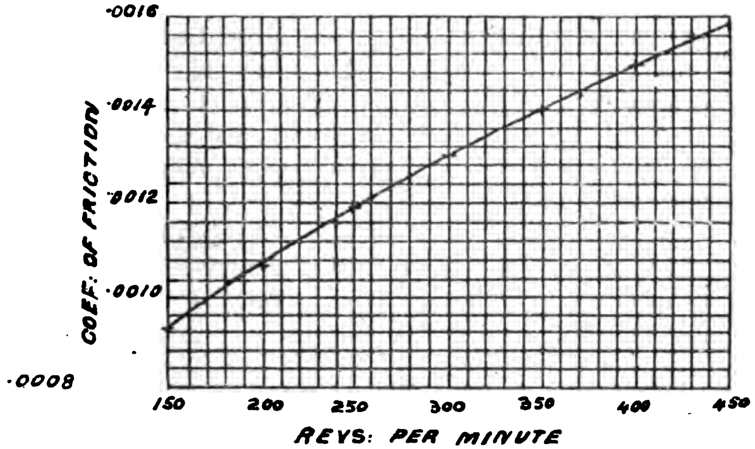


FIG. 6.

Thus, the load being practically doubled—520, instead of 258 lbs. per square inch—the co-efficient is halved, being ·001 instead of ·002.

CASE II.	Bath of mineral oil.	Revs. per min., 300,
	Load per sq. in., 100 lbs.	Co-eff. of fric., ·0062,
	" " 310 "	" " ·00207,

where, with different nature of oil and with much higher velocity, the co-efficient of friction is reduced by one-third when the load is practically trebled.

The second point to be noted is, that as the velocity, or number of revolutions, increases, the value of the co-efficient increases too. We have shown, for solid friction, that there is (see Fig. 4) a decided *decrease* in the value of the co-efficient as the velocity increases, while the theory of fluid friction requires that the value of the co-efficient should *increase* as the square of the velocity. Thus it will be seen how that compound friction follows in this matter the laws that govern fluid friction rather than those that govern solid friction. The curve plotted out in foregoing diagram (Fig. 6) shows this increase. It has been plotted from the tables of Mr. Tower, and shows that the increase is directly as the square root of the velocity.

The third point to be noted is the marked way in which the co-efficient of friction diminishes as the temperature rises. We append (Table IV.) a few results showing this.

TABLE IV.

Bath of Lard Oil. Load, 100 lbs. per sq. in. Revs., 100 per min.

Temperature, Fahr. ...	60	70	80	90	100	110	120
Co-efficient of Friction...	·0059	·0048	·0040	·0034	0029	·0026	·0024

It will be remembered that Captain Galton accounted for the diminution in the value of the co-efficient after the brakes had been in contact some seconds by reference to a rise in temperature due to the rubbing, his suggestion being well borne out by the above.

The fourth point to be noted is the way in which, with oil baths, the journal was entirely oil borne, the pressure of the film of oil between the journal and brass being easily measured by a pressure gauge. Mr. Tower found that, by taking the number of readings on various parts of the brass, the average pressure per square inch was exactly equal to the load per square inch on the brass. This conclusively proves that no friction took place between metal and metal. The pressure was greatest at the top of the brass on the off side (see Fig. 7), and amounted to twice the mean load per square inch on the journal.

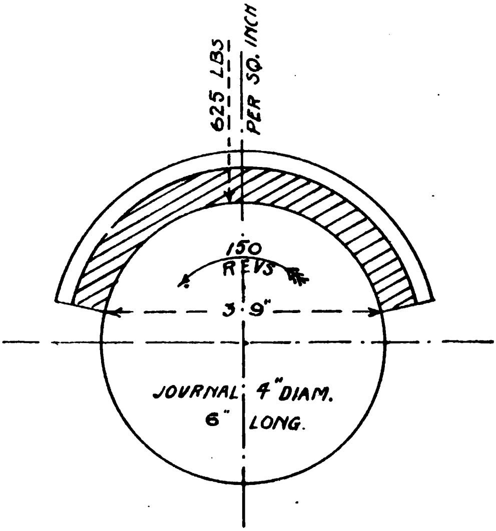


FIG. 7.

Experiments with Ordinary Methods of Lubrication.—In the experiments made with ordinary methods of lubrication *the first point to be noted* is that the co-efficient of friction was more nearly constant under varying loads, especially with oil pad under journal, and thus showed that the friction more nearly follows solid friction as insufficient lubrication is allowed.

The second point to be noted is that the co-efficient of friction does not show any decided tendency to increase with increase of velocity, as in case of oil-bath experiments, again following more closely solid friction.

The third point to be noted is that the part of the brass where the greatest pressure exists is the wrong place to introduce the lubricant, it being found that it was impossible to force it down between the journal and brass even when there was a head of seven inches of oil above the journal.

The fourth point to be noted is the proportion which the co-efficients of friction bear to one another when various methods of applying the lubricant are used. These are shown in the following table furnished by Mr. Tower. The lubricant was rape oil, the revolutions 150 per minute, and the load per square inch about 260 lbs.

	Co-efficient of friction.	Comparative values.
Oil bath...........	·00139	1
Pad under journal............	·00900	6·48
Syphon lubricator............	·00980	7·06

Generally speaking, we gather that with the most perfect method of lubrication the laws governing the friction are more nearly akin to those which apply to fluid friction; but that as the lubrication became less efficient, the laws of solid friction became more generally applicable. Mr. Tower remarked in the discussion which took place, that the most important practical lesson to be learnt from these experiments was, in his opinion, the value of profuse lubrication and the great diminution of friction obtained thereby. The greatest pressure per square inch of journal was 625 lbs. beyond this brass seized, the greatest velocity was 450 revolutions per minute; that is 471 ft. per minute, or 7·85 ft. per second, at the outside surface of the journal. The lowest co-efficient obtained was with the oil bath, using olive oil, when the pressure per square inch was 520 lbs., the revolutions 150 per minute (giving a velocity at the surfaces of 157 ft. per minute, or 2·61 ft. per second), as low a value as 0·0008 being then obtained.

When, however, the pressure upon a bearing is exerted at definite intervals, and then only for a short time, as in the case of the eccentric pins of punching machines, where a pressure amounting to two tons per square inch is usual at the moment of shearing, the results given above will, of course, not hold; the time during which the pressure acts being too short to squeeze out the film of oil, seizing or abrasion taking place when the metals come into direct contact.

We just add that in the above experiments, as is usual in engineering circles, the pressure per square inch on a bearing is calculated by dividing the total pressure upon the bearing by the product of the diameter into the length of the bearing; thus, if the load were 2,400 lbs., and the journal 4 in. diameter by 6 in. long, the pressure per square inch would be

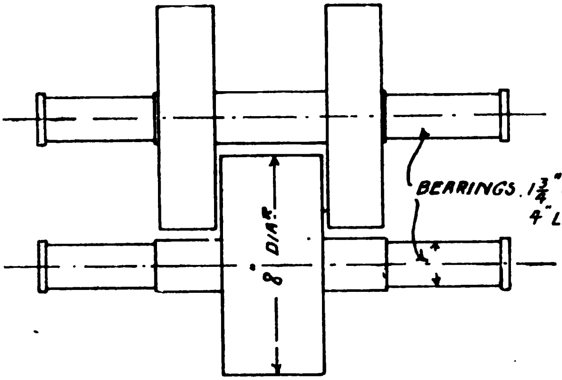
$$\frac{2{,}400}{4 \times 6} = 100 \text{ lbs.}$$

CHAPTER IV

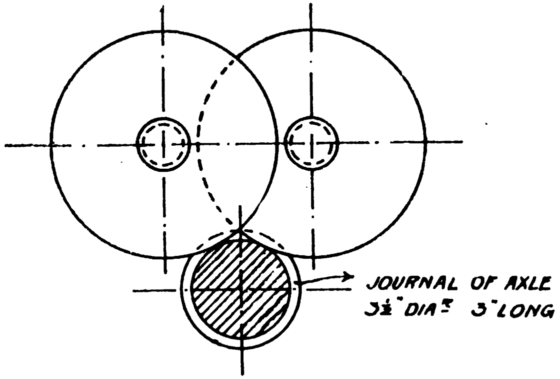
EXPERIMENTS ON FRICTION

Experiments of Mr. A. M. Wellington, Mr. C. J. H. Woodbury, and Mr. Goodman.

Experiments of Mr. A. M. Wellington.—Eighteen years ago Mr. A. M. Wellington read before the American Society of Civil Engineers a paper, giving a very interesting and instructive account of some experiments carried out by him, with the special object of determining the amount of journal friction at very low speeds. The chief thing before him was the friction to be overcome in starting from rest, in the case of railway journals, or what the author names *initial friction.* He had already—1878—made some experiments with actual loaded trucks and cars upon a gradient—generally known as the "drop test"—in order to ascertain this; the result being obtained by noticing the difference between the time theoretically required to reach certain fixed distances from the starting-point and the time actually taken. The experiments which formed the subject of the paper read in 1884 were made to corroborate those made by the drop test. The machine used was in many respects like that of Mr. Tower's (Fig. 5, page 28), but the load, instead of being applied directly as in Mr. Tower's, was applied through levers which multiplied a small load. The scale pan was dispensed with, the lever resting upon the table of an ordinary railway-station weighing machine; this was found to give in every



PLAN OF BEARING:



SECTION SHEWING AXLE:

Fig. 8.

way satisfactory results. The journal experimented upon was $3\frac{7}{16}$ inches diameter by 7 inches long, 10 revolutions per minute of which answered to that which would be produced by a train with the standard wheels moving at the rate of one mile per hour. Two methods of supporting the bearing were tried: first, an ordinary brass, such as is usual in axle boxes; and secondly, the Higley Patent Roller Bearing, where the axle runs upon two rollers, which are each supported upon two small bearings 1¾ in. diameter by 4 in. long. Fig. 8 gives an outline showing the way the journal or axle is supported, the brasses which carry the ends of the rollers not being shown.

The method of lubrication adopted was free lubrication by means of a pad. The lubricant was mineral oil. The comparison of the results of the two kinds of bearings is very interesting, as it gives us an insight into the behaviour of roller bearings, and their value in relation with ordinary journals. The experiments showed that the Higley bearing was very efficient for low speeds. In two experiments under the same conditions, viz. total load on journal = 13,439 lbs., and just starting from rest, the co-efficients were as 0·0245 is to 0·112, that is, the ordinary bearing four and a half times as difficult to rotate as the Higley; but with the same load, and at a velocity of 100 revolutions per minute, corresponding to a train velocity of nearly ten miles per hour, the co-efficients were as 0·011 is to 0·019, or as 1 is to 1·7, which is a very marked drop.

The two curves in Fig. 9 show graphically the results obtained under similar conditions, the top curve being obtained from the ordinary journal, and the bottom one from the Higley roller bearing. The curves give the impression that had the experiments been extended to higher velocities, there would have been no advantage on the side of the patent bearing. As we should expect, trucks fitted with these bearings can be pushed about the yard with great facility, and their adoption for tram-cars seems a move in the right direction, seeing that they are so constantly starting and stopping, and never move at any high velocities. The investigations of Mr. Tower did not touch at all on one of the chief ends to which Mr. Wellington's tests were directed—initial friction, as he calls it. His conclusions, somewhat abbreviated, made under five heads, were as follows:—

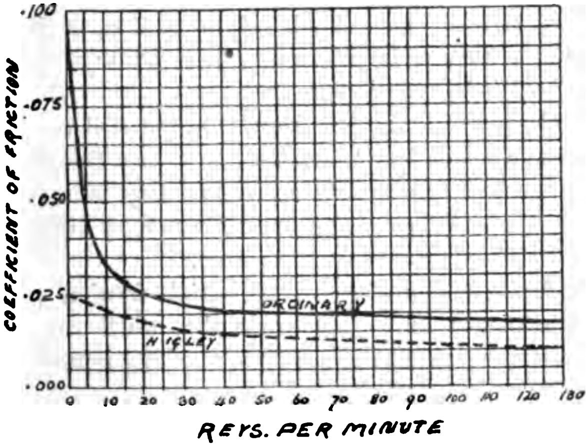


FIG. 9.

1. Friction at very low journal velocities of 0 + * is abnormally great, and more nearly constant than any other element of friction under varying conditions of lubrication, load, and temperature. It varies from 0·09

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* The symbol 0 + is used to denote a very low velocity—hardly appreciable in extent.

to 0·12 for loads of 30 to 280 lbs. per square inch. Within these limits it is not greatly modified by load or temperature.

2. This abnormal increase of friction is due solely to the velocity of revolution, continuing unchanged so long as the velocity is unchanged, and returning to the same amount whenever the velocity is reduced to the same rate. It is not appreciably affected by the fact that the journal may be just starting into motion, or is just coming to rest, or is temporarily reduced to a velocity of 0 + during continuous motion.

3. At velocities higher than 0 +, but still very low, the same general law obtains. The co-efficient falls very slowly and regularly as the velocity is increased, but is constantly more and more affected by differences of lubrication, load, and temperature.

4. A very slight excess of initial friction proper could generally be observed over that which continued to exist at the nearest approach to a strictly infinitesimal velocity which it was possible to obtain.

5. There is no such phenomenon in journal friction as a *friction of rest* or *quiescence* in distinction from (that is *differing in amount from*) *friction of motion* at slow velocities and due to the fact of quiescence. Consequently the use of such a term, although convenient, is scientifically inaccurate, in that it ascribes the phenomenon to a wrong cause, and to a cause which is not necessary for its existence.

His conclusions agree well with those we have been referring to in our last chapter, in which we pointed out that the investigations of Prof. Fleeming Jenkin tended to show that there was no abrupt change between static and dynamic friction, in fact no reason why we should introduce two terms, for without contradiction they were used as names for quantities which differed from each other in more ways than one.

Experiments of Mr. Woodbury.—Mr. C. J. H. Woodbury, of Boston, Mass., U.S.A., in the years 1880 and 1884, read two papers before the American Society of Mechanical Engineers, entitled "Measurements of the Friction of Lubricating Oils." The experiments which these papers described were conducted in the interests of the Mill Mutual Assurance Companies of New England. It may seem, at first sight, strange that underwriters should be interested in the properties of lubricating oils, but when we consider how many disastrous fires in cotton mills owe their origin to the use of indifferent oils, it will be seen that they have, in order to guard their own interests, to take advantage of all that the latest theoretical knowledge can suggest or ingenuity devise. It will be observed that these experiments were not to test journal friction, but the relative values of different oils for lubrication.

After trying nine different machines and not being able to obtain satisfactory results, Mr. Woodbury designed a special machine embodying those points that were found to be desirable, and yet not found combined in any one machine that he tried. It may be pointed out that the pressures per square inch of surface were very low, varying from 1 lb. to 40 lbs. per square inch, so that a machine well adapted for heavier work would not be sensitive enough in this case. It was found that the following must be known or preserved constant, in order to obtain a useful machine:—The temperature, the velocity, the area of the frictional surfaces, the thickness of the film of oil between the surfaces, and the mechanical effect of the friction.

The machine devised consisted of two annular discs rubbing together, the lower one of hardened tool steel

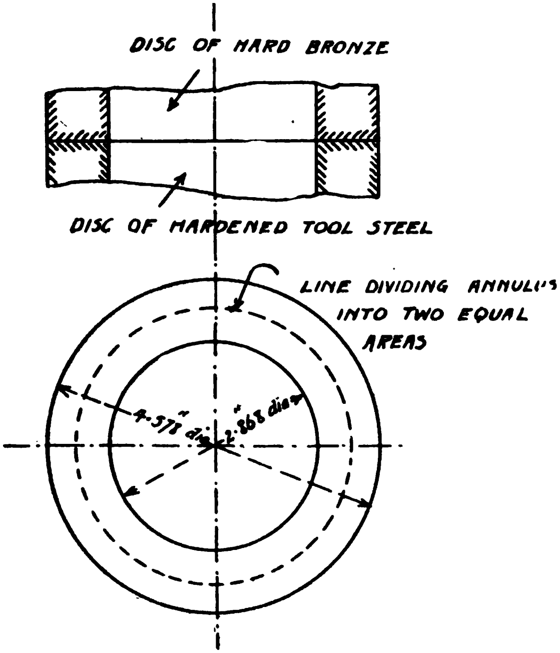
DISC OF HARD BRONZE
DISC OF HARDENED TOOL STEEL
LINE DIVIDING ANNULUS INTO TWO EQUAL AREAS
4·578" dia
2·868" dia

FIG. 10.

and the upper one of hard bronze, coated with a thin film of platinum. The outside diameter of the discs was 4·578 inches, the inside diameter being 2·868 inches. This encloses an area of rubbing surfaces of 10 square inches. The mean velocity of the rubbing surfaces was obtained by finding the length of that circle, which would just divide the annulus into two equal areas, as shown by the dotted line in Fig. 10. The diameters of the discs were so proportioned that this line was 12 inches long, so that the number of revolutions per minute gave in linear feet the mean velocity at which the top surface passed over the bottom one. Means were adopted so that no heat escaped by radiation, that the temperatures could be accurately ascertained, and that water at freezing point or other temperatures could be introduced so as to preserve even temperatures.

TABLE V.

FRICTION OF PARAFFIN OIL.—Velocity of Rubbing, 300 ft. per minute.—Flashing Point, 342 deg. F.—Specific Gravity, 0·888.

Pressure in lbs. per square inch.	Temperature, Fahr.			
	40 deg.	60 deg.	80 deg.	100 deg.
1	0·5380	0·3400	0·2120	0·1380
5	0·1400	0·0900	0·0620	0·0436
10	0·0836	0·0554	0·0402	0·0302
15	0·0633	0·0435	0·0325	0·0257
20	0·0528	0·0375	0·0289	0·0227
25	0·0460	0·0336	0·0265	0·0210
30	0·0413	0·0307	0·0246	0·0199
35	0·0376	0·0286	0·0231	0·0190
40	0·0349	0·0271	0·0221	0·0181

The above table gives the most important results of Mr. Woodbury's experiments, and careful consideration of them will verify what that gentleman observed, "that in a general way the co-efficient of friction diminishes inversely with the pressure and directly with the fluidity of the oil as indicated by the temperature (the greater the temperature, the more fluid the oil), and that the rate of these differences diminishes with the increase of pressure. The reason for this is that the resistance due to the viscosity of the oil is greater at low than at high temperatures, and that with heavier pressures the film of oil is actually thinner, besides being relatively smaller in proportion to the pressure."

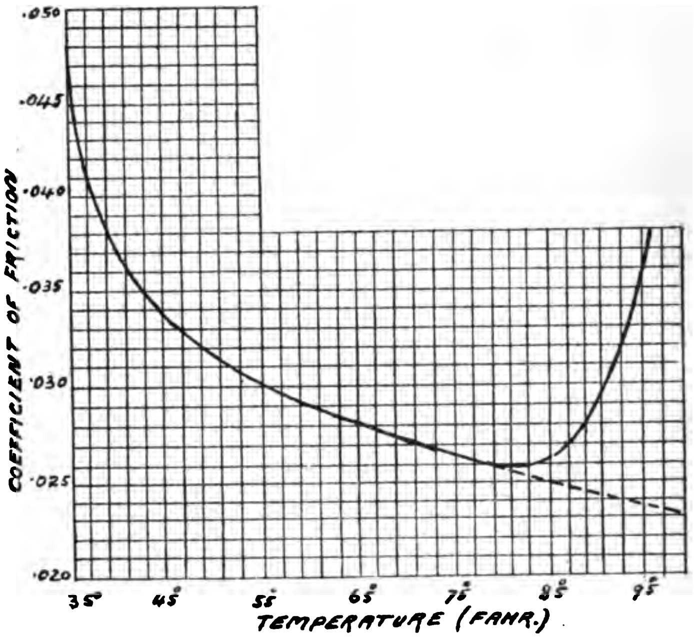

Fig. 11.

The way in which Mr. Woodbury explains the manner in which the films of oil act is worthy of notice. He says: "A film of the lubricant adherés to each of the frictional surfaces, and that portion which lies between these two films is pulled in one direction on one side, and in the other direction on the other side, and as a resultant, the movement of this centre layer is a rolling motion, whose rate of progression varies with the difference between the adhesion of the oil between the two frictional surfaces."

He also gives an account of the test made with a very limpid mineral oil, which, as soon as the temperature passed 80 deg. Fahr., was so incapable of withstanding the light pressure of 33 lbs. per square inch that it permitted the dics to touch one another, thus raising the value of the co-efficient by the addition of sliding to fluid friction. A very complete test record was given, from which the figures in Table VI. have been taken, and also from which the diagram, Fig. 11, has been prepared. The dotted line shows how a less fluid oil would have gone on with a reducing co-efficient as the temperature rose.

TABLE VI.

FRICTION OF PARAFFIN OIL (very limpid).—Velocity of Rubbing, 296 ft. per minute.—Pressure per square inch, 33 lbs.

Temp.	Co-eff.	Temp.	Co-eff.	Temp.	Co-eff.	Temp.	Co-eff.
F. deg.		F. deg.		F. deg.		F. deg.	
35	0·0485	50	0·0325	65	0·0288	85	0·0261
40	0·0371	55	0·0310	70	0·0275	90	0·0297
45	0·0344	60	0·300	75	0·0265	95	0·0370
				80	0·0260		

We have now to call our readers' attention briefly to **Experiments of Mr. Goodman**, which will bring this section to a close.

The experiments were carried out upon a machine similar to that of Mr. Tower's, but with several modifications introduced by Mr. Goodman. The bearing was 2 in. diameter, and the brass 4 in. long. One of the chief points to which attention was directed was "to find what relation exists between the total frictional resistance of a brass and the chord of its arc of contact with the journal." Lubrication was carried out under four different conditions—

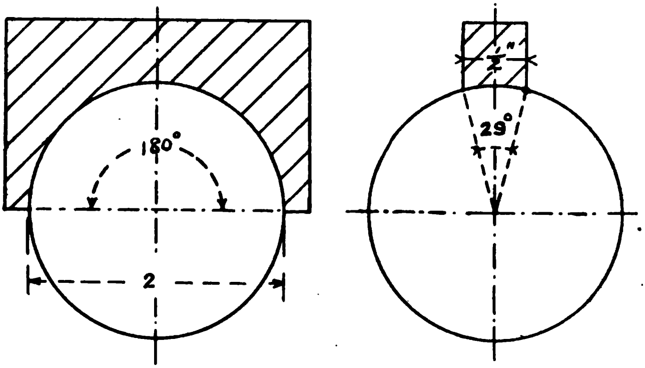

Fig. 12.

1. "Perfect, or oil bath," as described in Mr. Tower's paper.
2. "Satuated pad, in which a pad, standing in oil, was placed so that its upper part pressed on the bottom of the shaft, and which represents the maximum efficiency of pad lubrication."
3. "Ordinary pad, where a pad was used with a very meagre supply of oil, namely, 0·02 lb. of oil on 0·05 lb. of pad. This was found, by experiment, to be the smallest quantity consistent with safety, and represents the minimum efficiency of pad lubrication, the pad used in general practice coming between these limits."

4. "The syphon lubricator, provided with lamp cotton to feed three drops of oil per minute."

The chords of contact varied from 2 in. subtending an angle of 180 deg. to ½ in., which subtends an angle of 29 deg. Fig. 12 shows the two extreme cases.

Mr. Goodman found that the total frictional resistance was reduced by reducing the width of the brass. This will be seen by consulting the two following tables, which have been compiled by taking the results observed when the load on the brass was 300 lbs., which is about a mean value, for the loads varied by increments of 50 lbs., from 50 lbs. to 550 lbs.

Table VII. gives the results for 233 revolutions per minute, and Table VIII. for 15 revolutions per minute.

TABLE VII.

TOTAL LOAD ON BRASS, 300 lbs.—Revolutions per minute, 233.—Temperature, 40 deg. Cent.

Length of Chord of Brass in Contact.	2 in.	1¾ in.	1½ in.	1 in.	½ in.
Bath Lubrication ...	0 0103	0·0081	0·0071	0·0037	0·0028
Sat. Pad " ...	0·0143	0·0134	0·0080	0·0066	0·0035
Pad " ...	0·0285	0·0238	0·0196	0·0128	0·0094
Syphon " ...	—	—	0·0231	0·0150	0·0196

TABLE VIII.

TOTAL LOAD ON BRASS, 300 lbs.—Revolutions per minute, 15.—Temperature, 20 deg. Cent.

Length of Chord of Brass in Contact.	2 in.	1¾ in.	1½ in.	1 in.	½ in.
Bath Lubrication ...	0·0031	0·0024	0·0021	0·0016	0·0049
Sat. Pad " ...	0·0038	0·0031	0·0024	0·0015	0·0015
Pad " ...	0·0065	0·0042	0·0030	0·0019	0·0016
Syphon " ...	—	—	0·0115	0·0066	0·0595

By taking an average of all the figures given in the foregoing two tables we obtain an instructive comparison of the four different ways used for lubrication. Thus, considering the oil-bath method as unity, the others stand:—

Saturated pad	1·28
Pad with little lubricant	2·32
Syphon	4·8

It was to be expected that the diminution of the co-efficient of friction with the reduction of width of the brass would stop somewhere, and increase with any further reduction in the width of the brass, depending to some extent upon the method of lubrication, etc. This can be seen in Table VII. in the results opposite the syphon method, where the co-efficient increases from 0·015 for 1 in. width to 0·0196 in. for ½ in. width. Again, in Table VIII. there is a similar change when the brass is reduced from 1 in. to ½ in. wide in the case of bath lubrication, thus showing that a decrease in the number of revolutions and in the temperature affects the turning-point.

It is also noticeable, that at the lower rate of revolution the rate of change in the value of the co-efficient is not so marked as with the higher rate of revolutions either when the load varied or the width of the brass. This agrees with what Mr. Wellington pointed out about low velocities.

Mr. Goodman also made some experiments upon a ball bearing taken from a "Rudge" bicycle. It was composed of 11-ball $\frac{5}{16}$ in. diameter. The diameter of the groove in which the balls ran was 0·875 in.

The following laws appeared to hold good for the special bearing under consideration:—

I. The co-efficient of friction of ball bearings is constant for varying loads, hence the frictional resistance varies as the load.

II. The friction is unaffected by a change of temperature.

III. The co-efficient of friction with ball bearings is rather higher than that of plain bearings, when bath lubrication is used; but it is lower than when they are lubricated by the ordinary methods.

IV. They will also run easily with a much smaller supply of oil than will a plain bearing, hence there is a decided advantage in favour of ball bearings.

It is worth our readers' while to compare these results with those obtained from the Higley Patent Bearing.

The work done by Prof. Thurston has not escaped our notice, but although confirming what we have already brought before our readers, it deals more especially with the values of different kinds of oil for lubricating purposes, and will, consequently, find a better place when we come to that part of our subject. It is intended in our next chapter to commence the consideration of lubricants themselves, the various kinds, and the more common and easily applied tests for them.

CHAPTER V

OILS AND LUBRICANTS

Methods Adopted for the Reduction of Friction—Lubricants—Effect of Heat on Oils—Oxidation—Grit in Oils—Co-efficient of Friction.

Methods Adopted for the Reduction of Friction.—In order to form a correct judgment as to the most suitable lubricant to employ under any given set of conditions, it is essential that we should have a general idea of the different classes of oils and other lubricants in general use; not, of course, to obtain an exact knowledge of all the processes of manufacture, for that is a distinct industry in itself, and means that a person has to apply the greater part of his days to it. An outline of the comparative value of lubricants, both commercially and in connection with lubrication, and also a few simple methods of testing their properties and detecting any adulteration, must be expected in any who would desire to attain economy in lubrication. Of course, it is a very easy thing to ask for a set of rules to guide, but the true way, as we see it, is to understand the first principles of the matter, and then, with the help of those rules or suggestions which long and special acquaintance with the subject can afford, to select that which will be the most advantageous to use. In this, as in every other kind of knowledge, some small amount of trouble is necessary in order to become acquainted with a general idea of the question, and then a slight application of common-sense in order to put two and two together, and arrive at a conclusion, as in every case the lubricant must be selected with care and regard to the work it has to do.

Lubricants.—These may be divided, for convenience, into three classes:—

Liquid,
Semi-solid,
Solid.

The two latter classes are not used so largely as the first class, which embraces the oils in general use. Under the head of solid lubricants come plumbago or graphite and soapstone, with various patent compositions of which one or other of the above forms the principal ingredient. Semi-solids are those compositions, such as axle grease, which are solid at ordinary temperatures, but become more fluid as the bearing heats slightly. These will be considered at greater length when we have taken a rapid survey of the first class.

Liquid Lubricants or Oils may be divided into three distinct classes in keeping with the three great kingdoms in nature, from which they are severally derived.

I.—Animal Oils,
II.—Vegetable Oils,
and III.—Mineral Oils,

to which may be added a fourth, viz.—

IV.—Compound Oils,

artificially obtained by the combination of mineral with either animal or vegetable oils. Of these, animal and vegetable oils are more costly than mineral oils, but up to within some few years they were almost exclusively used for lubrication; but since the utilization of petroleum has made such immense strides, the mineral oils, either pure or compounded, as indicated above, have rapidly taken the leading place in the market.

Animal Oils naturally divide themselves under two heads, (1) animal oils proper, and (2) fish oils. The principal ones in common use derived from animals are—

Lard and Lard Oil,
Neatsfoot Oil,
Tallow and Tallow Oil,
and Bone Oil.

Those that come under head (2) are—

Sperm Oil,
Whale Oil,
Seal Oil,
and Cod Liver Oil.

Among these sperm oil is, without doubt, the best lubricant, especially for light work, both in its friction-reducing qualities and its power of endurance, or life. For heavy work it is not so suitable as some of the heavy vegetable and mineral oils. It is hardly necessary to add, that it is by far the most expensive oil in the market, so that its use is restricted in great measure. Neatsfoot and lard oils come directly after sperm, and, though inferior to it (see Table IX.), are yet excellent lubricants, and are used more extensively than any other of the animal oils. Tallow is also a good and much-used lubricant.

Vegetable Oils.—Of these the principal oils in every-day use are—

(*a*) Linseed Oil,
Hempseed Oil,
Poppyseed Oil,
Nut Oil,

(*b*) Olive Oil,
Palm Oil,
Rapeseed Oil,
Colza Oil,
Almond Oil.

Those under the head (*a*) are commonly known as drying oils, on account of their readiness to absorb oxygen from the atmosphere, especially when in thin layers, and become gummed, a property of considerable value for the purpose of paint making, but which renders them of no value as lubricants.

The list on the right-hand side are called non-drying oils, as they exhibit very little of the above-mentioned tendency, and, consequently, are free for the purposes of lubrication. Olive oil is the best vegetable lubricant; it ranks equally with lard or neatsfoot in its lubricating properties, its commercial value being about the same as those.

Between these two divisions come castor and cotton-seed oils, which cannot be termed non-drying, and yet are not so rapidly gummed as to be entirely worthless as lubricants. Cotton-seed oil is, or was, much used in the United States of America; and castor oil, on account of its great "body," is still often used for heavy machinery, such as the shafting of screw-ships, though some consider it far too thick for even the heaviest machinery in this climate. If used it should be fresh.

Mineral Oils are generally divided into two classes: *shale oils*, obtained by the distillation of bituminous shales, such as are found in Scotland and elsewhere, cannel coal and coal tar. To these may be added the *distilled oils*, obtained by distillation and subsequent condensation from the residues obtained in refining petroleum for illuminating purposes. The oils after distillation are subjected to chemical treatment with acid and alkali washing, which chemical treatment, as will be afterwards explained, it is very difficult to remove every trace of. These oils, having a good pale appearance, are often attractive, but the treatment they undergo reduces their body and lowers their flashing point, of which more anon.

The other class, *natural oils,* are obtained chiefly from the United States of America, although some come from the oil fields of Russia, and need little or no treatment to fit them for the purposes of lubrication, thus giving rise to their name. They have been divided into three classes by Mr. Brannt, of America, as follows; we give his words:—

1. *Natural oils requiring no treatment.*—The amount of oil suitable for general lubrication without some manipulation is very small in comparison with the quantity used for that purpose. Pure natural West Virginia oil of a specific gravity of ·880 is suitable for all kinds of heavy machinery, and will remain limpid in cold climates.

2. *Natural oils requiring only exposure to the air for the removal of adhering light vapours.*—Crude oils of specific gravity ·875 and under are brought into tanks partially filled with water and exposed to the action of the sun and air from twenty-four to forty-eight hours, according to the quantity of volatile oils present.

3. *Natural oils reduced by steam heat in open vessels.*—The oil is brought into tanks partially filled with water, and provided with coils of steam pipes and heated to 110 deg. Fahr. By this arrangement oil of about ·864 specific gravity is reduced in two or three days to about ·880 without any impairment of quality.

Natural oils having undergone no chemical treatment are much more suitable for use in cylinders, and, in fact, upon any metallic surface, whether heated or not.

Properties of Lubricants.—The properties characteristic of good lubricating oils must now be considered as well as those features which, while found in many oils, render them practically useless for our purpose, and which, therefore, it is desirable should be absent as far as possible, or, if not entirely absent, to a large extent neutralized.

Body.—The first property to be considered is the body of the oil, often termed its "viscosity" or "fluidity." This is really, in plain language, the thickness of the oil, and is generally measured by the number of *seconds* of time which it takes a definite quantity to pass through an aperture so small that it can only pass through in drops. A standard graduated glass vessel being used for this purpose, a relative value can be obtained and used for comparison with other oils. Means are also provided in order that the "body" may be determined at any desired temperature, which is very important in the case of cylinder oils, which must maintain their "body" at the temperature of the steam admitted to the cylinder. Mr. J. Veitch Wilson, chief of the lubricating oil department of Price's Patent Candle Company, who has made a special study of this subject, made some experiments with various oils in order to find whether thick or thin oils gave the best results in reducing friction and heat in machinery. In Table IX. we reproduce his results. The last column shows the average temperature the bearing ran at, and gives a measure of the oil's value in reducing friction. It will be remembered that we have pointed out in a previous chapter that the friction of a lubricated bearing consists of two parts—the fluid friction of the lubricant, and the friction due to any contact there may be between the bearing surfaces of the metals. The values given below embrace both of these, although the predominating part will be the fluid friction of the oil.

TABLE IX.

Name of Material.	Spec. grav. at 60 deg. Fah.	Body, in Seconds, at			Heat Developed.
		60 dg. F.	120 dg. F.	180 dg. F.	
Water	1·000	—	—	—	Fahr.
Castor Oil ...	·960	—	132	41	158 deg.
Resin Oil ...	·990	—	—	—	155 deg.
Engine Tallow	—	Solid	41	26	—
Tallow Oil ...	—	143	37	25	141 deg.
Neatsfoot Oil ...	—	112	40	29	—
Rape Oil ...	·916	108	41	30	148 deg.
Lard Oil ...	·916	96	38	28	146 deg.
Olive Oil ...	·915	92	37	28	143 deg.
Sperm Oil ...	·880	47	30	25	133 deg.
Mineral Oil ...	·905	45	—	—	121 deg.
" " ...	·875	30	—	—	117 deg.

The lubricants are placed in the order of their "body" at 60 deg. Fahr., and it will be noticed that with one exception—tallow, the body of which alters rapidly with the temperature—the average temperature was lower, showing the friction was less, as the "body" of the oil became less. Mr. Wilson concludes from these experiments that "the thinnest oil that will keep the surfaces apart is the best and most suitable for any bearing, and that extra thickness means a higher temperature and greater driving power." This agrees with what Prof. Thurston says:—"In order that any substance may be efficient as a lubricating material, it must possess (1) enough "body" or combined capillarity and viscosity to keep the surfaces between which it is interposed from coming in contact under maximum pressure;

(2) the greatest fluidity consistent with the preceding requirement—*i.e.* the least fluid friction allowable. The fluidity of the oil is not only different in different cases, but is very variable with change of temperature." He (Mr. Wilson) found, however, that the mineral oils, whilst possessing great fluidity, yet were soon exhausted, and the bearing ran dry, showing that though fluidity is desirable as giving less drag on a bearing, yet a certain amount of body cannot be dispensed with, since it is necessary to secure endurance in the lubricant. The animal and vegetable oils, on the other hand, are by no means deficient in this important respect, and are, on this account, often mixed with mineral oils to impart body to them. The best oil for this purpose is sperm, although lard, neatsfoot, and olive oils are in general usage. Thus the best properties of both kinds are secured.

Acidity and Decomposition.—It is highly desirable that all lubricants should be free from any ingredient that would attack and cause the corrosion of metals. Such corrosion is caused by the presence of acids. These may be present from two causes—(1) natural free acids, and (2) from traces of the acid which has been used in refining the oil. There is another cause which may give rise to their presence, and that is when a vegetable or animal oil suffers decomposition. It is well known that animal and vegetable oils consist of glycerine, in combination with stearic and oleic acids. In the manufacture of candles from animal and vegetable fats it is necessary to separate the glycerine from the stearic and oleic acids, and this decomposition is brought about "by the action of high pressure steam in the presence of water, generally with, but sometimes without, the addition of a small proportion of a base like lime." It will be at once seen that the conditions necessary for the decomposition of such fats and oils —as, for example, tallow—exist exactly in the interior of a steam cylinder, and whenever suet or some other animal or vegetable fat or oil is in requisition as a lubricant, it must inevitably become decomposed or reduced to its simpler elements. The greater the temperature in the cylinder, the sooner will this come about, and, in fact, for gas engines the use of such materials for lubrication is out of all question. Mineral oils are entirely free from this serious drawback, and do not suffer decomposition when exposed to heat, but are distilled into vapour without leaving any residue behind. In the preparation for use, too, of mineral oils there is not the same necessity for the use of any acid to effect their purification, and in the case of those best fitted for cylinder lubrication no acid is used at all. The mineral oils in this respect are very valuable, since if, as is often done, it is better to introduce some animal or vegetable oil, as before mentioned, to give body or endurance to a mineral oil, the percentage of that which would cause corrosion is reduced in a very great degree when compared with the use of a vegetable or animal oil alone.

There is another point of advantage which mineral oils have over the other two classes, and that is that it has no tendency to develop an acidity with age. We have it, however, upon the authority of a distinguished chemist,* that even if animal and vegetable oils be entirely free, when first manufactured, from any trace of acid, they may acquire a sensible acidity in time.

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* Mr. A. H. Allen, of Sheffield.

Effect of Heat on Oils.—The effect of heat on oils is divided for us into four parts, according to the peculiarities exhibited as different temperatures are reached. They are—

Volatility,
Flashing Point,
Firing Point,
Point of Solidification.

Volatility.—This is the tendency of an oil to evaporate when exposed to the atmosphere. It may be looked at from two standpoints, both important: first, as to the danger incurred on account of the risk of fire, due to the saturation of the surrounding atmosphere with combustible vapour, and then from the point of view of economy; since a lubricant that loses a considerable percentage of its volume by evaporation in a short time is not so economical as one that does not exhibit this feature in such a marked degree, though it may be somewhat more expensive. Evaporation is of importance only in connection with mineral or compound oils, as vegetable and animal oils do not show any tendency in that direction, and more so as the use of mineral oils with very low flashing points is very extensive in compound oils. Many of the mineral oils are given to evaporation, that is, are volatile, at the ordinary temperature of the air; how much more so when used in a hot engine-room, or upon bearings whose tendency is always to run a little hot! Prof. Ordway, of the Massachusetts Institute of Technology, in a report made by him for the N. E. Cotton Manufacturers' Association, U.S.A., gives some account of different oils tested for evaporation by exposing them to a temperature of 140 deg. Fahr. for twelve hours. He found the loss to vary from 1 per cent. to as much as 25 per cent. Out of one hundred and sixty-four kinds tested by him, he found what is tabulated below :—

TABLE X.

Amount of Loss	1%	5%	10%	15%	20%	25%
No. of Samples	11	48	46	31	14	4

That is, scarcely 7 per cent. were really good in this respect. However, the conclusion cannot be avoided, that were there no demand for cheap oils they would find no sale, so that the fault is not alone on the side of the manufacturer, but also on the part of those purchasers who consider nothing but cheapness. Oils can be obtained which lose nothing when exposed to the severer test of ten hours at a temperature of 212 deg.—that of boiling water ; but one that stands the above temperature for six hours without loss may be considered a first-class lubricant. This is a most important feature to determine in the case of cylinder oils. There should therefore, in testing such, be a difference made as to the temperature. Thus for cylinder oils and those intended for internal lubrication, a test has been proposed of an exposure to a temperature of 360 deg. Fahr. for six hours, and in the case of oils intended for ordinary purposes to 212 deg. Fahr. for the same period.

Flashing Point.—This is the temperature at which an oil gives off vapour in sufficient quantity to ignite when a lighted taper is brought near to it. This is accompanied with a little flash, hence the name "flashing point." This, again, is of importance in connection with mineral oils, animal and vegetable oils being very safe in this respect, the temperatures at which they flash varying between 400 deg. Fahr. and 540 deg.

Fahr., olive and sperm oils occupying the lowest place, whilst refined cotton-seed and lard oils are about the best. It is with mineral oils that danger is to be feared, since some low-class oils give off dangerous vapours at as low a temperature as 80 deg. Fahr. The generally acknowledged standard in this country is 350 deg. Fahr., those flashing below 300 deg. Fahr. being considered unsafe for general use. For cylinder oils a much higher temperature is required, Mr. J. Veitch Wilson giving 500 deg. Fahr., which gives a good margin of safety, since the temperature of steam at 200 lbs. per square inch is a little over 380 deg. Fahr. The methods by which this property, and the others too, is determined we shall deal with later on.

Firing Point is that temperature at which an oil *itself* will ignite and burn. It is very patent that this will be much higher than the flashing point, so that if this latter is kept up to the standards above indicated, there need be little fear for firing point. Mr. J. Veitch Wilson is disposed to attach more importance to evaporation than to either flashing or firing points.

Point of Solidification.—This feature of an oil is of importance when the machinery upon which it is used is exposed to the changes of our climate. If the lubricant should become set or even thicken, considerable wear might take place in bearings from the metals coming into direct contact through the cessation of the supply of oil. Prof. Thurston gives summer sperm as thickening at about 65 deg. Fahr., freezing at about 50 deg. Fahr.; winter sperm 50 deg. Fahr. and 35 deg. Fahr. Lard oil begins to harden at 40 deg. Fahr., solidifying at 7 deg. below the freezing point of water, 32 deg. Fahr. The heavier mineral oils freeze at 12 deg. below freezing point of water, viz. 20 deg. Fahr., while the lighter ones remain fluid at 0 deg. Fahr. Some obtained from the Russian oil fields exhibit a very low freezing or solidifying tendency. The mineral oils as a class stand in a much better position in this respect than either of the other two classes.

Oxidation.—The property of oils to, in greater or lesser degree, absorb oxygen from the atmosphere, we shall consider under two heads—(1) Gumming; (2) Spontaneous ignition.

1. *Gumming.*—One effect of the absorption by an oil of oxygen is that it thickens, or becomes resinous or gummed. We have already alluded to this when pointing out the distinction between drying and non-drying oils. Those most affected by this are the vegetable oils; of the animal oils, those derived from fish are all affected in this way with the exception of sperm, where the gumming tendency is very little. The animal oils proper (lard, neatsfoot, tallow) are not affected in this way to any appreciable extent. Mineral oils, however, stand foremost in this respect, being entirely free from the tendency to absorb oxygen when in contact with the atmosphere. Sperm oil, for instance, if exposed to the temperature of 140 deg. Fahr. will gain in weight two or three per cent. in twelve hours. It will be seen at once how objectionable this feature is in a lubricating oil, since, as it becomes more and more sticky, it will throw a correspondingly extra amount of work on the engine in order to drive the machinery. Then again, it necessitates more frequent cleaning of bearings, etc., to say nothing of the waste of power and time in starting when the machinery has rested at week end.

2. *Spontaneous Ignition.*—In addition to "gumming" there is also another dangerous element due to oxidation. Whenever an oil prone to oxidation is diffused through a quantity of cotton or other waste, it is always more or less liable to spontaneous ignition due to the heat developed in the process of rapid oxidation under these peculiar circumstances. This is rendered more dangerous still if it should be near any heated body, as a steam or hot-water pipe. Nothing is more common than to see waste, saturated with oil, thrown down in some corner of a shop or mill on Saturday midday, without the slightest regard to the danger incurred thereby. There is not the slightest doubt that it has been the occasion of many otherwise unaccountable conflagrations. Mineral oil, since it does not absorb oxygen either alone or in contact with cotton waste or other fibrous material (jute, say), is entirely free from this baneful feature. Mr. Galletly conducted some experiments to ascertain the time a certain amount of cotton waste soaked with different kinds of oil took, under favourable circumstances, to ignite spontaneously. The oil waste was placed in a paper box and put in a hot-air bath, which was maintained at a temperature ranging between 130 deg. and 170 degs. Fahr. It will be seen from the figures given below that oils differ much from one another in the time required to ignite.

TABLE XI.

Oil.	Time.
Boiled Linseed ...	1 hour 15 mins.
Seal Oil	1 „ 20 „
Raw Linseed	4 „ 0 „
Lard Oil	4 „ 0 „
Gallipoli Olive ...	5 „ 0 „
Refined Rape	9 „ 0 „

But when equal parts of mineral and seal oil were experimented upon, the waste refused to ignite. The late Mr. J. J. Coleman, F.C.S., repeated these experiments, desirous of studying more particularly the effect of mineral oil in the retardation or entire prevention of spontaneous combustion. The temperatures between which Mr. Coleman's experiments were made were higher than those used by Mr. Galletly, being 180 deg. and 200 deg. Fahr. instead of 130 deg. and 170 deg. Fahr. The following are the figures given by him :—

TABLE XII.

Oil.	Time.
Seal Oil on Wool	3 hours 0 mins.
Whale Oil on Wool	3 „ 15 „
Whale Oil on Cotton	3 „ 0 „
Whale Oil on Jute	9 „ 0 „
Olive Oil on Cotton	4 „ 0 „
Oleic Acid on Wool	4 „ 0 „
Olive Oil, with 20 per cent. of Mineral Oil ...	8 „ 0 „

The following mixtures refused to ignite after twenty-six hours' exposure to the above-mentioned conditions:—

Equal parts of olive and mineral oil on cotton.
„ „ whale and mineral oil on jute.
„ „ seal and mineral oil on wool.
„ „ oleic acid and mineral oil on wool.
Oleic acid, with 20 per cent. mineral oil on wool.

We thus see that the addition of mineral oil to animal and vegetable oils exercises a very powerful effect upon oxidation, and is thus another great point in favour of mineral and compound oils.

Professor Ordway, in the report already alluded to, gives the results of some experiments made since those given above, and has arrived at practically the same results. In his case he took a given weight of cotton waste, about a handful, and impregnated it with its own weight of the oil to be experimented upon, paying attention that the oil was equally spread throughout. He says: "We have made experiments with cotton waste and cotton-seed oil mixed with petroleum-mineral oil, and have found that cotton-seed oil, mixed to the amount of 25 per cent. with 75 per cent. of petroleum oil, will take fire spontaneously; so it seems that although spontaneous combustion is retarded in a great degree, it is not entirely prevented, even by a pretty large admixture of mineral oil, in the case of such oil as cotton-seed and linseed, which are peculiarly liable to oxidation. When we came to take lard oil. a careful experiment was made, which showed that 33 parts of petroleum oil mixed with 67 parts of lard oil would not undergo spontaneous combustion at the temperature at which the experiment was made; whereas with 52 per cent. it did undergo spontaneous combustion. We know cotton-seed oil is one of the oils we have to fear, and cotton-seed oil happens to be one of those whose spontaneous combustion cannot be prevented by a slight admixture of petroleum-mineral oil. But the experiments of Dollfus and Coleman were correct, it seems; we had no reason to doubt that they were correct, but the experiments we made were made at a little higher temperature; and although the oil, mixed in the proportion of 70 parts of oil and 30 of petroleum, may not take fire when the temperature is maintained at 110 deg. Fahr., yet it may when it is maintained at 190 deg. Fahr.; and of course cotton waste is liable to be exposed sometimes to a steam heat, and a steam heat may range up to 300 deg. Fahr. Still, it is a fact that the admixture of even 10 per cent. of one of the heavy petroleum oils does diminish very much the tendency to oxidation or to spontaneous combustion, and is a fact, of course, of immense importance." We must not confound flashing tendency with spontaneous ignition, they arise from different causes; the first is characteristic of mineral oils, while the latter is characteristic of animal and vegetable oils.

Grit in Oils.—Any oil containing grit in it is, obviously, very expensive to use, since it conduces so much to wear and tear. Mineral oils are in this respect often offenders, since the oil in rising in the petroleum wells gets charged with earthy matters, which may or may not be removed in the process of manufacture. Many thick black oils, too, that are sold cheaply consist of the "still bottom" or residuum produced in refining illuminating oils, so that care should be exercised in this direction. A simple test, given by Prof. Thurston, is to drop a little of the oil on clean white blotting paper, which absorbs the oil, leaving the impurities visible as black specks on its surface.

Co-efficient of Friction.—We have already, in Chapters II. and III., referred to fluid friction, and given the laws that govern it; this is an important factor when we have under consideration the value of a lubricant, and we remind our readers of what we then said, that compound friction is a combination of solid and fluid friction, and we shall be readily followed when we say, that as the conditions of lubrication more nearly approximate to either of these two kinds, so the results will show that the frictional resistances are governed by the laws which are observed to act in that kind of friction to which it approximates. So that to gain any information on the friction of an oil we must go to experiments conducted under the condition that solid friction was eliminated as much as possible, leaving that due to fluid friction as the main factor.

Compound or mixed friction most nearly approaches fluid friction when a journal is flooded or entirely oil borne, as in the case of the "Oil-Bath Experiments" of Mr. Tower. The following tables give some figures deduced from his data, to which we add those deductions which appear most important:—

TABLE XIII.

100 lbs. pressure per square inch. 100 Revolutions per minute, equal to 105 lineal feet per minute.			100 lbs. pressure per square inch. 450 Revolutions per minute, equal to 471 lineal feet per minute.		
Nature of Lubricant.	Co-efficient of Friction.	Relative Figure.	Nature of Lubricant.	Co-efficient of Friction.	Relative Figure.
Sperm Oil ...	0·0025	**46**	Sperm Oil ...	0·0064	**42**
Rape Oil ...	0·00277	**51**	Rape Oil ...	0·00714	**48**
Mineral Oil ...	0·00334	**62**	Mineral Oil ...	0·0078	**52**
Lard Oil ...	0·0035	**65**	Lard Oil ...	0·009	**60**
Olive Oil ...	0·0054	**67**	Olive Oil ...	0·0089	**59**
Mineral Grease	0·0054	**100**	Mineral Grease	0·0151	**100**

N.B.—Bearing 4 in. diameter, 6 in. long.

TABLE XIV.

520 lbs. pressure per square inch. 150 Revolutions per minute, equal to 157 lineal feet per minute.			520 lbs. pressure per square inch. 450 Revolutions per minute, equal to 471 lineal feet per minute.		
Nature of Lubricant.	Co-efficient of Friction.	Relative Figure.	Nature of Lubricant.	Co-efficient of Friction.	Relative Figure.
Sperm Oil ...	Seized	—	Sperm Oil ...	Seized	—
Rape Oil ...	0·00095	**68**	Rape Oil ...	0·00148	**67**
Mineral Oil ...	0·00123	**88**	Mineral Oil ...	0·00186	**85**
Lard Oil ...	0·0009	**64**	Lard Oil ...	0·0017	**78**
Olive Oil ...	0·0008	**57**	Olive Oil ...	0·0017	**78**
Mineral Grease	0·0014	**100**	Mineral Grease	0·0022	**100**

N.B.—Bearing 4 in. diameter, 6 in. long.

Deductions.—(1) That the pressure per square inch upon the brass remaining the same, the relative value of the lubricants does not alter appreciably when the revolutions are increased from 100 to 450 per minute, answering to velocities of 105 and 471 lineal feet per minute at the surfaces in contact. (2) That the difference in the relative value of the oils is very greatly reduced as the pressure per square inch is increased from 100 to 520 lbs. per square inch, the difference in their lubricating powers being less at the high pressure. (3) That sperm, though having the least co-efficient of friction, has also the least weight-carrying power. (4) That lard and olive oils differ very slightly from one another under all the above conditions. (5) That the mineral oil and mineral grease carried the highest load, 650 lbs. per square inch, although nearly approached by rape oil, which seized after 573 lbs. per square inch.

TABLE XV.

625 lbs. pressure per square inch.

Revs. per minute.	150	200	250	300	350	400	450
Bath of Mineral Oil ...	0·0013	0·00139	0·00147	0·00157	0·00165	—	—
,, ,, Grease	0·001	0·0012	0·0014	0·0014	0·0016	0·0018	0·002

The above table shows the superiority of a thick lubricant over a thin one at a high pressure, and that although an increase in the number of revolutions increases both their co-efficients, yet the ratio of them is not varied much.

Mr. Tower says that the mineral *oil* was a thinner sample than the mineral *grease;* the former being fluid at 50 deg., while the latter could only be described as grease at 50 deg. This will account for the oil experiments showing less friction than with the grease, except with the highest load, at which, the thin oil being overloaded and on the point of seizing, the friction is greater than with the grease.

The above figures relate to fairly high pressures, 100 to 625 lbs. per square inch, so we now give some data as obtained by Mr. Woodbury, U.S.A., at low pressure, moderate speed, and standard temperature, *the conditions being as nearly as possible those met with in spinning frames.* The following table gives his figures, the relative figures being added by us, as before in Mr. Tower's, the worst being taken as 100.

TABLE XVI.

CO-EFFICIENT OF FRICTION FOR SPINDLE OIL.

Order of Value.	Nature of Lubricant.	Co-efficient at 100 deg. F.	Relative Figures.
9	Refined Petroleums, Heavy Spindle...	0·1187	**49**
12	,, ,, Heavy Spindle...	0·1233	**51**
10	,, ,, Heavy Spindle...	0·1208	**50**
4	,, ,, Light Spindle ...	0·1113	**45**
5	,, ,, Light Spindle ...	0·1132	**46**
1	,, ,, Extra Machinery	0·0756	**31**
2	Bleached Winter Sperm	0·0956	**39**
3	,, Winter Sperm	0·1067	**44**
6	,, Winter Sperm	0·1141	**47**
7	Unbleached Winter Sperm	0·1147	**47**
8	Bleached Winter Sperm	0·1170	**48**
11	,, Winter Sperm	0·1217	**50**
13	Seal Oil...	0·1608	**66**
14	Lard Oil	0·2181	**89**
15	Neatsfoot Oil	0·2427	**100**

From these figures we see that, under certain conditions, neatsfoot oil stood in the relation of

$$\frac{100}{31} = 3{\cdot}2$$

to the best refined light petroleum oil; that is, it absorbed over three times as much power; while the best winter sperm stood in the relation of

$$\frac{39}{31} = 1{\cdot}26$$

to the same oil; that is, it absorbed one and a quarter times as much power as the best refined light petroleum.

It will be seen at a glance, upon comparing the value of the co-efficients in Tables XVI. and XIV., how very much greater the power required to drive is in the case of light machinery than in the case of heavy machinery; and so it is more essential, in the case when light machinery forms the plant of a manufacturer, to attend to the co-efficient of friction of the oil he uses than when heavy machinery is in use.

In all the experiments above referred to the lubricant was pure, whether animal, vegetable, or mineral oils were used; we now turn to some in which mixed or compound oils were used. Prof. Thurston, U.S.A., referring to some of his experiments, says:—"The mixing of mineral and animal oils yields, in some cases at least, unexpected results. Thus a mineral oil, rich in paraffin, being compared with lard oil gave the following under 100 lbs. pressure per square inch:"—

Mineral oil	0·0150 co-eff. of friction.
Lard oil	0·0160 ,, ,,
Mineral oil 95 parts, lard oil 5 parts ...	0·0120 ,, ,,
Mineral oil 90 parts, lard oil 10 parts...	0·0140 ,, ,,

While for relative endurance, putting the mineral oil as 100, he obtained—

Mineral oil	100
Lard oil	120
Mineral oil 95 parts, lard oil 5 parts	125
Mineral oil 90 parts, lard oil 10 parts	160

This shows unmistakably the value and decided superiority of the compound oils under the conditions set forth.

We now give a brief epitome of some results published in *Engineering*, April 20, 1888, where they can be examined *in extenso*. They were carried out by Mr. J. Veitch Wilson, to whom we have already had occasion to refer, in conjunction with Mr. T. J. Pullin, of Messrs. Bass & Co., a gentleman who has invented a special machine for oil testing, to which we may refer later on in our articles, our desire being now to deal with the *results* obtained from it. Two sets of experiments were made, the first with various standard oils commonly used for lubrication—what we have already termed pure oils; and secondly with various compound oils used for lubrication, these last being the manufactures of the company to which Mr. J. Veitch Wilson belongs. These experiments showed conclusively that properly prepared compound oils run lighter—that is, have a lower co-efficient of friction—and also run cooler than any simple animal or vegetable oil of like gravity and body.

The pure oils experimented with were—

Sperm	Southern and Arctic.
Olive	Gallipoli, Smyrna, and Saffi.
Neatsfoot	British.
Lard Oil	American.
Tallow Oil	British.
Refined Rape Oil	"Special," German and English.
Castor Oil	French 1st.
Tallow (solid)	British.
Colza	British.

The compound oils experimented with were mixtures designed to fill the place of the pure ones given above.

TABLE XVII.

Nature of Lubricant.	Specific Gravity.	Body at 60 deg. Fahr.	First Series of Tests.				Second Series of Tests.			
			Velocity of rubbing surfaces in ft. per minute.	Pressure in lbs., total load on brass.	Working Temperature.	Drag.	Velocity of rubbing surfaces in ft. per minute.	Pressure in lbs., total load on brass.	Working Temperature.	Drag.
Southern Sperm	881	51	1110	60	189	18	459	160	114	28
"Supermoline"	879	50	1110	60	181	15	459	160	110	24
Castor Oil	964	—	1110	60	132	41	459	160	158	69
"Polar Oil X"	934	234	1110	60	208	27	459	160	182	43

The bearing upon which these experiments were made was 3 in. diameter by 3⅛ in. long, giving an area of 9·375 square inches. This gives for the first series 6·4 lbs. per square in. on the brass, and for the second series 17·07 lbs. per square inch, which are much lower pressures than Prof. Thurston's; they are nearer to those we quoted of Mr. Woodbury's. The velocities, it will be noted, are very high. It is much to be regretted that the "drag" was not expressed in coefficients of friction instead of the arbitrary way in which they are, since we have no means of comparing them with the results of other investigators. The lubricant was supplied through an oil cup freely. It is at once seen, from Table XVII., that, with a light oil, as sperm, the corresponding compound placed immediately below it, "spermoline," is a better lubricant, working with less drag and a lower temperature under both the tests; the same is observed in a heavy oil, "castor oil," and the substitute advised for it. Nothing is said of the relative endurance of pure and compound oils in the paper, the test not being directed to that end; but we think that, had any been made, there is no reason to suppose that they would have shown different results from those we have already brought before our readers' attention, which were carried out by Prof. Thurston, showing that compound oils were not deficient in that important respect.

CHAPTER VI

PROPERTIES OF OILS AND TESTING SAME

Manner of Testing Properties of Lubricants—Tests for Body—Redwood's Viscosimeter—Tests for Acidity and Decomposition—Tests for Volatility—Tests for Flash and Firing Points—Gray's Flash Point Apparatus—Test for Firing Point—Test for Point of Solidification—Test for Gumming—Bailey's Apparatus—Test for Spontaneous Ignition.

Manner of Testing the Properties of Lubricants.—We have briefly, but yet we hope clearly, brought before our readers the properties of lubricants, and we intend to give a short account of some tests for same, the majority of which can be, with very little trouble, carried out by machinery users. It is well to have the desire to test the lubricants one uses, but those desires must be directed into scientific channels. We have heard on good authority of an old gentleman who utilized the back of his store-room door as the first and final means of testing oils. When oils were to be examined a few drops of each of the oils were poured upon the upper part of the door, and the rate and extent of their progress downwards was carefully noted, but what was deduced from the data thus obtained we have not been able to learn. Sometimes a person exhibits a scientific turn of mind, and his methods are not less instructive than those of the rule-of-thumb gentleman just referred to. The scientific man pro-

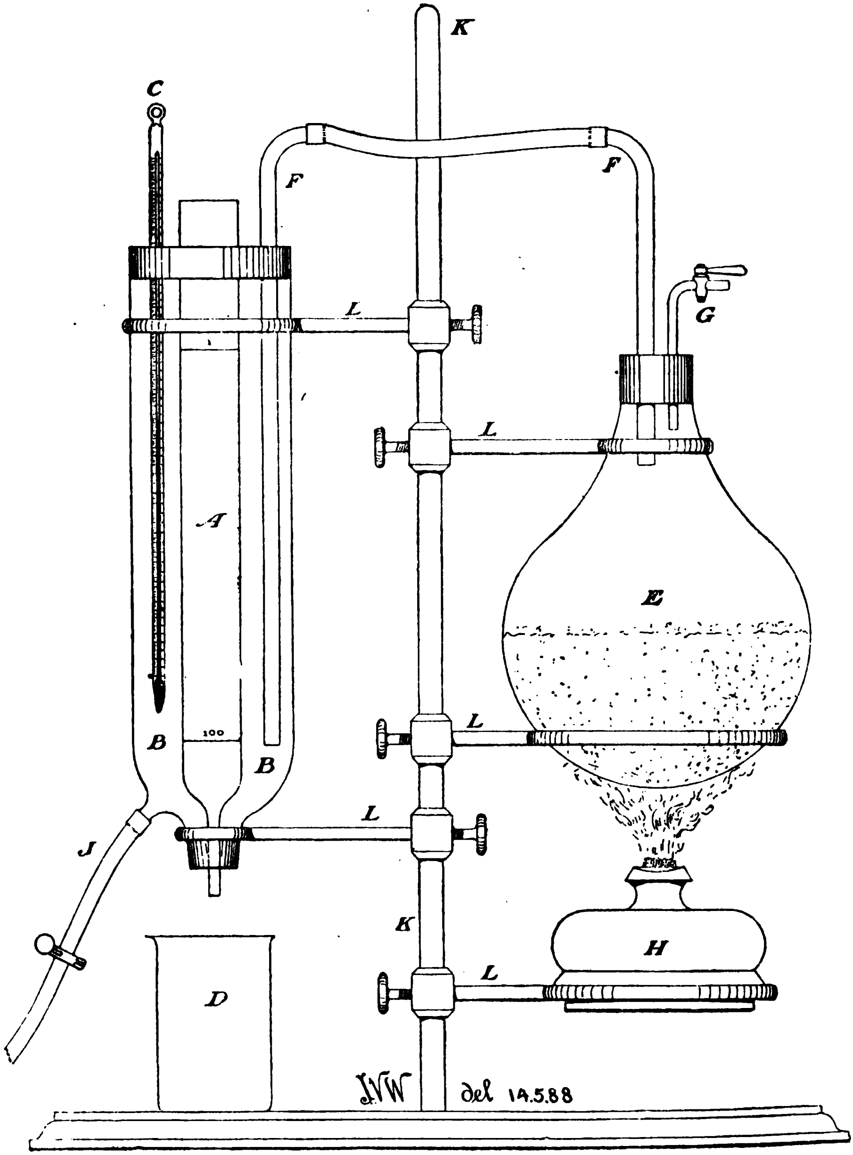

FIG. 13.

vides himself with an oleometer or a hydrometer, and, if he is very scientific, with some primitive form of viscosimeter. The fact that he does not understand these instruments or the way to use them seems to aid greatly the faith he reposes in them, and the satisfaction that he derives from their occasional use.

Tests for Body.—The first we come to is "body" or "viscosity." This is tested by a viscosimeter; there are several of these used, constructed on various principles.. The one shown in the accompanying illustration (Fig. 13) is one based on the principle of permitting the oil to pass through a small aperture, and expressing the body of the oil by the number of seconds it takes for a definite and standard quantity to pass through this aperture. The apparatus, we believe, was primarily the design of the late Mr. J. J. Coleman, but we understand that various little improvements by Mr. J. Veitch Wilson have been added to give it its present form.

Referring to the illustration, A is the glass tube into which the oil to be tested is placed; it is 1 in. diameter and about 15 in. long. It has two marks upon it; the first up to which the oil is filled, and the second the mark at which the time taken is noted. It is surrounded by a glass jacket, B, B, which is full of water, the temperature of which can be regulated by the admission of steam by means of the pipe F, F, from the flask E; C is a thermometer, the bulb of which is in the water of the jacket B, B; J is an indiarubber pipe with a spring clip so that water may be let out from the jacket vessel in case of its becoming full through the condensation of the steam; G is a small cock to relieve the flask of extra steam if need be; D is a jar into which the oil drops; H a spirit lamp to heat the water, and L, K, the stand. The following method of using the apparatus is given by Mr. J. Veitch Wilson:—"The oil to be tested is brought to the required temperature in a glass or tin vessel, the water in the jacket being maintained at the same temperature; the oil is then transferred to the inner tube (A), which is filled to the higher mark, the outlet being closed by the operator's finger, or, when oils are being tested at high temperatures, by a small wooden plug. At a given moment the finger or the plug is removed, the oil flows into the receiver (D), and the time required for the surface of the oil to fall from the higher to the lower mark is taken as the equivalent of the viscosity of the oil."

Prof. Thurston remarks of this test for oil that "*large consumers of oil* sometimes purchase on *the basis of this test solely.* It is regarded as quite as satisfactory and reliable as any single physical or chemical test known, *and as second only to the best testing machine methods.*"

It can easily be seen that by the use of the above apparatus the "*body*" can be determined up to as high a temperature as 212 deg. Fahr., which we explained was necessary for cylinder oils in the last chapter.

There is one objection, however, to the principle upon which this form of apparatus is based, and that is, the specific gravity of the oil enters into the question, and gives a *greater head* in the case of heavier oils, thus making them appear to have a slightly less body than what they really possess. In the case of two oils of near about the same specific gravity, the difference would not be very appreciable; but in the case of, say, sperm and castor oils, whose specific gravities are widely different, it makes itself felt.

Fig. 14 illustrates a form of apparatus[1] designed by Mr. James Napier, of Glasgow, to obviate this objection. The viscosity is measured by the resistance of the oil to the revolving part, and is determined by the period of time taken to perform a given number of revolutions under the influence of a weight C. A jacket E is also provided for obtaining any desired temperature. We are given to understand that the apparatus, as now made, has no annular spaces. It is merely a drum revolving in the oil. This alteration does not affect the principle of its construction, which removes entirely the factor of specific gravity—that is, the relative weights for equal volume of various oils.

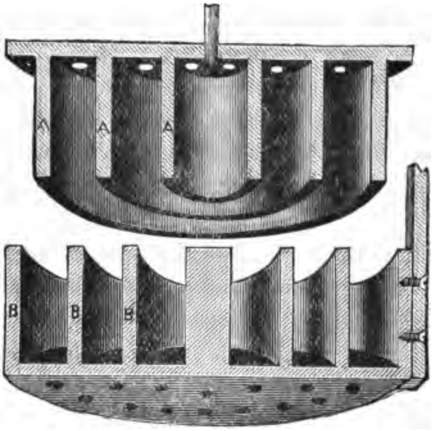

Fig. 14.

A simpler form of apparatus embodying the same principle was described in *Engineering,* February 9, 1894, as being in use on the Philadelphian and Reading Railway, U.S.A. A cylinder 2 in. diameter by $1\frac{1}{2}$ in. long is suspended in the oil to be tested by means of a fine steel wire attached at its upper end to a knob inserted in a stiff frame. By twisting this knob the wire can be subjected to torsion. About the middle of its length the wire has attached to it horizontally a

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[1] The illustration is from Redwood's *Petroleum* (Griffin & Co.).

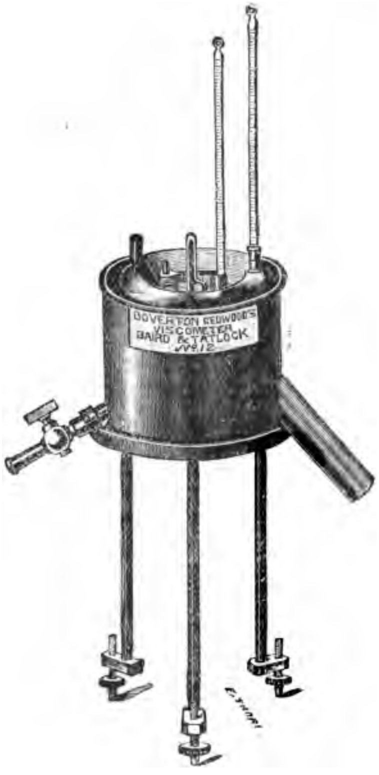

FIG. 15.

disc, graduated in degrees, which rotates with the wire when the latter is turned, and serves to measure the angles of oscillation. In making a test the oil is surrounded by a bath of paraffin or water to bring it to

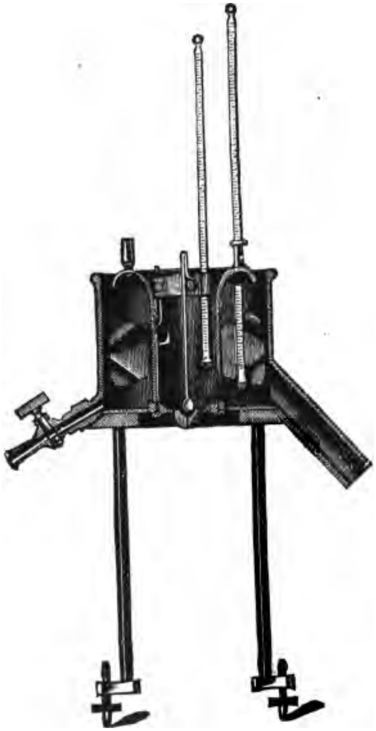

Fig. 16.

the required temperature. The disc on the wire is clamped, and the knob at the top turned through an angle of 360 degrees. The clamp holding the disc is then released, and the torsion of the wire causes the cylinder immersed in the oil to move through angles which become less at each succeeding oscillation. The angles are measured by the disc, and form a criterion of the viscosity of the oil. The above is very simple, and commends itself to us, but we have not met any one that has had practical experience with it.

Redwood's Viscosimeter.—This instrument owes its existence to the late Mr. W. H. Hatcher, but it has been improved by more than one, especially by Mr. Boverton Redwood, F.I.C., F.C.S., who improved it to such an extent that it is now generally adopted as a standard both in this and in foreign countries. It is plain from Figs. 15 and 16 that its principle of action is the same as that shown in Fig. 13. It differs from it in the following details. Copper is substituted for the glass, which was very liable to fracture; the reduction in depth of the receptacle which contains the oil to be tested, thereby reducing the effect of "head" due to difference in specific gravity; the use of a piece of agate for the jet, which lends itself more easily by drilling to secure orifices of more uniform size than is possible in glass; the projecting tube, closed at the lower end, under which a lamp is placed, providing a better method of heating the bath; and, last, the addition of a revolving agitator in the bath or jacket for uniformly distributing the heated liquid rising from the projecting tube. The oil cylinder has a stopper, consisting of a small brass ball attached to a wire, which closes the hole in the agate mouthpiece. Inside the oil cylinder, and at a short distance from the top, is fixed a small bracket, terminating in an up-turned point, which forms a gauge of the height of the oil level. Under the orifice is placed a graduated glass vessel, which denotes the volume of oil passing (vol. ii. page 194) from the oil cylinder. Mr. Allen, in his *Organic Analysis*, vol. ii. page 194, before referred to, gives an illustration of an improvement of his own which gives *a constant head.* It is extremely ingenious, but space forbids our reproducing it.

To use the apparatus, the bath is filled with a suitable liquid; water answers well for temperatures up to 200 deg. Fahr., and for higher temperatures a heavy mineral oil may be used. The liquid having been raised to any desired temperature, the oil to be tested (also heated) is poured into the oil cylinder until the top of the gauge is reached. A narrow-necked flask, holding 50c.c. to a point marked on the neck, is placed beneath the jet in a vessel containing liquid at the same temperature as the oil. The ball valve is raised, a stop watch started at the same time, and the number of seconds occupied in the discharge of 50 c.c. noted, which will give the viscosity of the oil in relation to some known or standard oil. Care should be taken that the oil is filtered before being tested, and also that the agate jet is free from the slightest trace of dirt. The apparatus should be cleaned thoroughly every time it is used by means of benzine and alkali, which will remove any trace of an oil used in a former experiment. There are other forms of viscosimeters, as Tagliabue's, Stillman's, and Perkins'; but we have given a good idea of this class of instruments, and, in conclusion, give the preference to Redwood's, as being generally accepted both in scientific and commercial circles.

Tests for Acidity and Decomposition.—(1) The acid due to refining. The presence of mineral acid, generally sulphuric, as before spoken of, may be detected by boiling a quantity of the oil to be tested with distilled water and dipping into the aqueous solution *blue litmus paper*, which will be turned red if any acid is present. (2) Free fatty acid may be determined by dissolving a known weight of the oil in a mixture of alcohol and petroleum spirit, to which a standard solution of alkali is slowly added by dropping, in presence of phenolphthalein, which is colourless when the solution is acid or neutral, but becomes pink when the acid is neutralized and the alkali is slightly in excess. Mineral acids are seldom present, but the free fatty acids are usually present in animal and vegetable oils, even to the high percentage of ten or more. Animal oils may be distinguished from vegetable oils by the use of the fact that chlorine turns animal oils brown and vegetable white.

Tests for Volatility.—The amount of volatility is determined by exposing a known weight of the oil in a small Berlin basin or watch glass in the hot chamber of a water bath to a temperature of 212 deg. Fahr. for a given number of hours, and noting the loss in weight. An oil losing more than 5 per cent. in ten hours should be rejected when we bear in mind that oils can be obtained which lose nothing, or a fraction of 1 per cent., in that time. A good cylinder oil should be equally satisfactory when tested for six hours in a sand bath at the temperature of 360 deg. Fahr.

Tests for Flash and Firing Points.—Flash point can be determined by the " open " and the " close " test. In the open test a glass beaker, about 1 in. in diameter by 2½ in. deep, being filled with the oil under examination, is placed in a sand bath, and carefully screened from draught, the temperature being raised gradually, 5 deg. to 10 deg. Fahr. per minute, during which a very small flame from a taper is passed over it at definite intervals until the vapour given off is in sufficient quantity to produce a slight explosion accompanied by a blue flash.

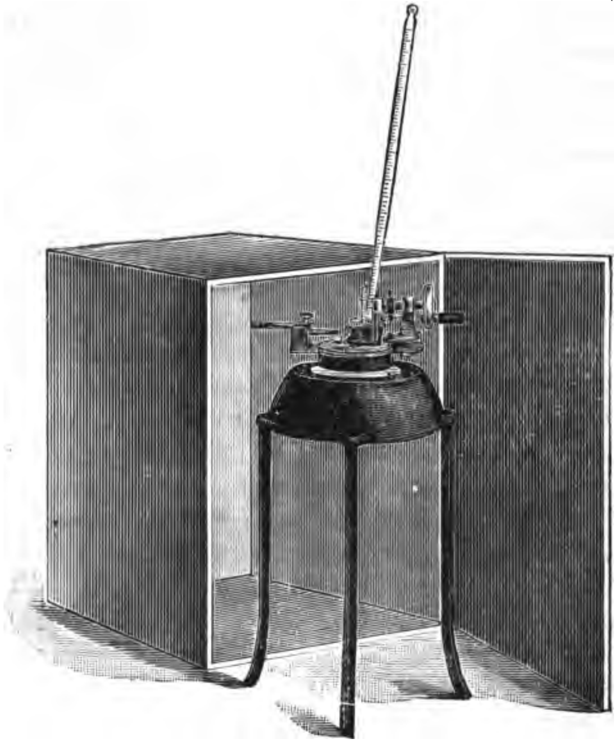

FIG. 17.

A thermometer whose bulb is inserted in the oil will give the temperature.

Gray's Flash Point Apparatus.—This instrument (Fig. 17) is designed especially for the heavier mineral oils, and is the best and most reliable instrument now offered to the public. It is based upon "Abel's Government Tester"; but various details have been modified to suit the different conditions under which it is used, Abel's instrument being used for burning oils only which have a low flash point. The cup A, which contains the oil to be tested, is the same size as that specified by Act of Parliament, and adopted in Abel's instrument, viz. $2\frac{2}{10}$ in. in diameter, and having a depth of $\frac{2}{10}$ in.; but instead of the oil being heated through the medium of a surrounding jacket containing water, the heat is applied directly underneath by a lamp, not shown in the accompanying figures. On this cup, which is set in a suitable tripod stand, is placed a tight-fitting lid; through the lid a spindle passes, which has inside the cup two sets of stirrers, one below the surface of the oil, and the other in the vapour space above. A horizontal spindle, suitably supported, is used to rotate the vertical spindle by means of the handle B and the bevel wheels G and H. The test is made by depressing the small flame of the test light D through a shutter into the oil chamber below, which produces a flash when sufficient vapour is given off by the heated oil. A thermometer is placed conveniently in the oil chamber to give the necessary readings. It will be seen from the figures and the foregoing description that the apparatus is a "close test" one.

To make an experiment, the makers advise that the oil cup A is filled up to the mark inside, the cap replaced, and the thermometer inserted in the tube provided for it. After lighting the heating lamp and placing it underneath the cup A, light the test light, adjusting the flame to about $\frac{1}{8}$ inch in length. During the heating of the oil the stirrers should be moved at short intervals by turning the handle B several times. The faster the oil is being heated the oftener the stirrers should be rotated. When nearing the point at which the oil is expected to flash, it is advisable to reduce the rate of heating in order that the observations may be more readily and accurately made. If necessary a rough test may be first made to give some idea of the temperature at which the oil is likely to flash. Tests may be made with the flame at every degree rise until the oil gives a blue flash. The rate of heating should not be too rapid, to allow of readings being taken at every degree rise. Redwood says that a fall of 1 in. of mercury in the barometer will lower the flashing point about 2 deg. Fahr. Since different forms of instruments give results somewhat divergent, it is necessary when comparing results to know what kind of instrument has been used.

Test for Firing Point.—The test for firing or burning point is usually that described as the open cup test for flashing point, the test being continued until the oil actually itself ignites and continues to burn, the temperature then noted being the burning point. If the flash point is satisfactory this test is not generally applied.

Test for Point of Solidification.—The following test for solidification is taken almost verbatim from Mr. A. H. Allen's *Organic Analysis*, vol. ii.:—A test tube, about 5 in. in length by ⅔ in. diameter, is fitted with a collar of cork or indiarubber by which it is fixed in the mouth of an empty flask. The oil is then poured into the tube till it is about two-thirds full, and a delicate thermometer is suspended freely in the liquid, so that the bulb may be wholly immersed. When the oil commences to solidify at the bottom of the tube the temperature must be carefully observed. The operator then stirs the contents of the tube slowly by giving the thermometer a circular movement, first three times to right and then three times to left. The first effect of the agitation is to cause the temperature to fall slightly, but afterwards a sensible rise takes place, and the mercury remains stationary for at least two minutes. The temperature thus indicated is the solidifying point of the oil, and the results obtained are remarkably constant. A freezing mixture can easily be made by any of the ordinary means, or water whose temperature is gradually lowered by the addition of iced water.

Test for Gumming.—The oxidizing or gumming tendency of oil may be found out by exposing small quantities in shallow iron cups or watch glasses at ordinary temperatures to the action of the atmosphere for ten days or a fortnight, taking care to exclude all dust. If more rapid results are necessary, the cups or watch glasses may be exposed to a water or sand bath at, say, 212 deg. Fahr., when ten or twelve hours will probably be enough time to indicate the character of the oils. This test is purely comparative, and so an oil known to be of good quality should be placed in one cup for purposes of comparison.

Bailey's Apparatus consists of an inclined glass plane (see Fig. 18). This plate is fixed in a metal box, which is constructed to receive a copper vessel which is filled with boiling water, and heats the glass. The box being lined with felt, the radiation is small, and the drop in temperature consequently slow. Six or less drops of different kinds of oil are started, as shown in the accompanying illustration, at the same moment from the top end; the time each takes to reach the lower end giving a comparative value of its gumming tendency. This instrument can also be used for testing the body or fluidity of an oil.

Test for Spontaneous Ignition.—This, as we have already explained, depends upon the action of the atmosphere being promoted by a higher than normal temperature. As it is desirable to expose as large a surface of oil to the atmosphere as possible, it is customary to conduct these experiments by saturating a small quantity of cotton-wool (say enough to fill a small wooden match-box) with the oil to be tested. The wool, after having been wrung out, is placed in the hot-air chamber of a water bath at a temperature of 212 deg. Fahr., the temperature of the wool being noted from time to time, till the mass shows a tendency to spontaneously ignite, or give evidence that it will not do so.

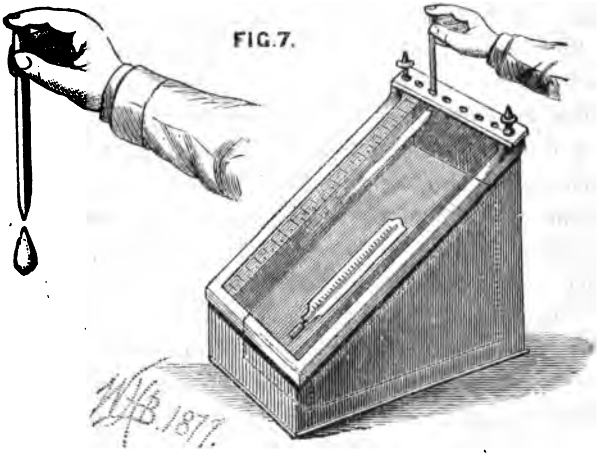
FIG.7.
WHB 1877.

Fig. 18.

CHAPTER VII

TESTING MACHINES

Thurston's Testing Machines—Prof. R. H. Smith's Journal Friction Testing Machine—Boult's Patent "Cylinder Lubricant" Tester—Ingram and Stapfer's Oil Tester.

It is very clear that the best possible means of testing the efficiency of anything is by placing it under exactly the same conditions that will obtain in the actual use it is to be put to. But it is often the case that this is inconvenient, and several of the conditions under which it will ultimately work have to be altered, as, for instance, in the case of a large compression member for a bridge, where it would be impossible to test it *full size*, and a small scale model is made use of instead, and the results obtained from the testing of that are multiplied in such a manner as to give a result for the actual member. This is of course open to a little error, but it is fortunate for machinery users that, generally speaking, they are not compelled to have recourse to a model for the test of lubricants, but can obtain tests from machines capable of being adjusted to give conditions, both as to velocity, diameter and length of bearing, pressure per square inch, and temperature which may be necessary. The tests given in the last chapter, though both necessary and useful in their place, must yield the palm in point of accuracy to those obtained by the machines we are now about to refer to. The great point, however, in their favour is that the tests can be carried out very much more quickly and at considerably less expense than by the use of testing machines, which means considerable outlay in the first place, and also power to drive them when in use. With large consumers of oil, such as railway companies, this initial outlay is justifiable, but for small consumers we consider that the tests given in the last chapter are all that is necessary, and should any special case arise, resort may be had to a laboratory possessing such a machine. It may then be asked why we refer to testing machines, but we consider this work would not be complete without a short reference to them, and consequently we shall not entirely omit them from our consideration, our desire being to refer after that to the oils placed upon the market by various manufacturers, and to point out the suitability of the grades they turn

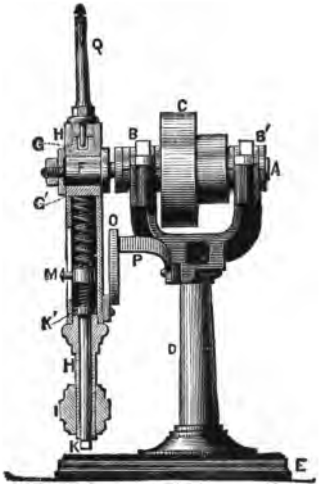

Fig. 19.

out for different purposes. Of the machines brought before the notice of our readers, the cheapest is Ingram and Stapfer's patent, made by Messrs. W. H. Bailey and Co. In a letter they received from a very well-known and eminently respectable firm who had purchased one of these machines, it is stated that it had saved them some hundreds of pounds. Where a firm's oil bill runs into anything like hundreds a year, a machine for testing should certainly be invested in, since it means then perhaps 3 or 4 per cent. only on their annual expenditure, and will enable them perhaps to effect a saving of from 10 to 30 per cent.

Professor Thurston's Testing Machine.—This machine is the invention and design of Professor Thurston, U.S.A.; the sole makers for Europe being Messrs. W. A. Bailey and Co., Limited, Albion Works, Salford, Manchester, to whose kindness we are indebted for permission to reproduce the wood-cuts illustrating these machines. The oil to be tested is placed upon the journal F (see Fig. 19), which in the machine adapted for general work is 1¼ in. diameter. The journal is carried upon the overhung end of the shaft A; this shaft is carried upon the two journals B, B′, firmly attached to the base E by the standard D. The pulley C, for driving the machine, is stepped in order to give variation in speed. A counter, not shown in Fig. 19, is placed at end of shaft to give the number of revolutions. Two gun-metal bearings G, G′ are fitted around the journal, and forced against its surface by the spiral spring immediately underneath. This spring is made adjustable by means of a screw worked by a spanner upon the nut K, and the pressure per square inch can be ascertained by the position of the pointer M upon an index plate.

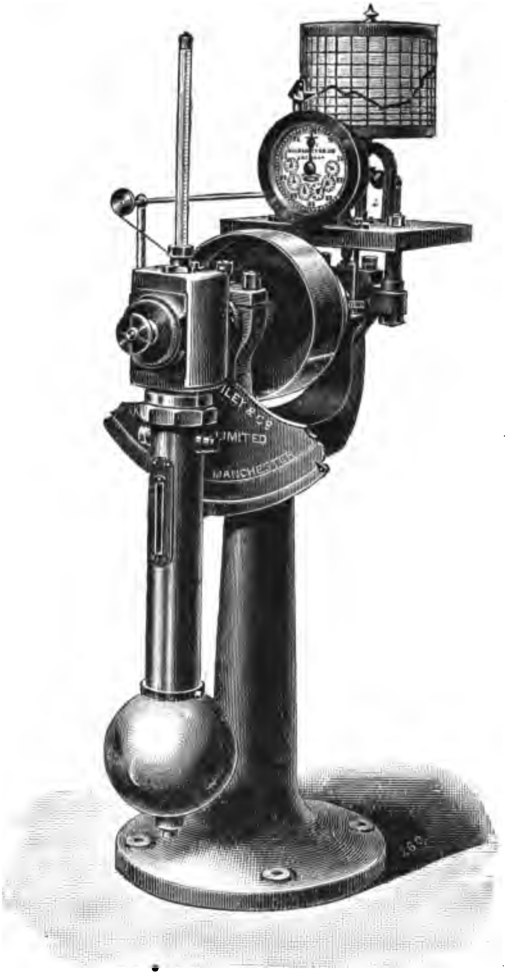

Fig. 20.

The brasses, spring, etc., are carried by a casing, H, in the form of a pendulum, to the bottom end of which a heavy ball, I, is attached of just sufficient weight to allow the arm of the pendulum to swing up into a horizontal position when the bearing is running under maximum friction; *i. e.* when the bearing is perfectly dry. A pointer moves along the arc P (which can be more distinctly seen in Fig. 20), and shows the angle from which the pendulum is deflected from the perpendicular. This arc is carefully laid off in such divisions that, dividing the figure the pointer is on by the pressure shown by the other pointer, M, gives the coefficient of friction at the time being. A thermometer, Q, is also attached to give the temperature of the journal.

In Fig. 20 we have a much larger pattern of the same machine, the journal for testing the lubricant being $3\frac{3}{4}$ in. diameter by 7 in. long, a size adopted to exactly correspond with certain railway axles. The total pressure which can be exerted by the spiral spring upon the brasses is 5 tons, thus giving a pressure of over 400 lbs. per square inch. A recording apparatus is fixed to this machine, which automatically registers the various readings on the graduated arc, the paper being arranged to advance $\frac{5}{16}$ in. for every 1,000 revolutions of the journal. The vertical height of the line drawn from the base line of paper gives the reading on front arc, or how much the pendulum swung out from perpendicular at any given time during a trial. This is, we need scarcely add, a splendid machine, and has been adopted by various railway companies and large oil users. In making permanent records of trials, five things should be entered:—(1) The pressure per square inch on the brasses; (2) the number of revolutions per minute; (3) the temperature at the time being; (4)

the readings on the arc of the machine; and then another (5) is added by calculation, which is the co-efficient of friction, and obtained by dividing the number in (4) by that in (1).

Professor Robert H. Smith's Journal Friction Testing Machine.—This machine, in use at Mason's College, Birmingham, is in our judgment a distinct improvement upon Prof. Thurston's, which we have just described.

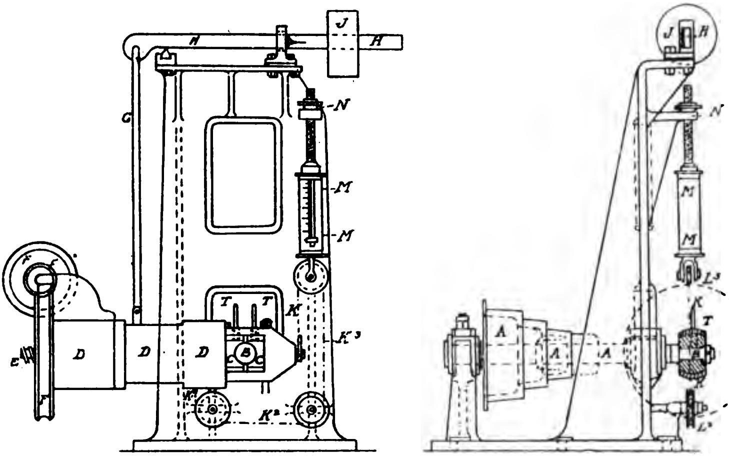

FIG. 21.

While the pressure is put upon the brasses by means of a strong spiral spring, and is in this respect the same as in Thurston's apparatus, yet the whole weight of the spring and its casing is taken from off the journal, and no pendulum weight being used, there is not a higher pressure upon the top than on the bottom brass as in Thurston's. A glance at the accompanying illustration shows how all the weight is held by the suspension rod G. This again gets rid of a feature somewhat objectionable in Thurston's machine, that is the tendency the journal has to break off at its neck, due to its having to entirely support the weight of pendulum, spring, etc. We do not for a moment question that this has been thoroughly thought of by the manufacturers and provided against by giving ample strength to all parts, but we refer only to the principle of the thing.

Referring for a moment to the illustration (Fig. 21) kindly lent to us by Professor Smith, B is the test journal rotated by means of the pulley A, A, A; this journal is made to screw into the end of shaft, thus allowing the introduction of different size journals; C, C are the bearings which are made to bear upon the journal with any desired pressure by means of spring enclosed in the barrel D, D, D; the spring being compressed and released by the worm and worm wheel F, F, F. Two thermometers, T, T, dipping into mercury pools in the two half bearings give the temperature of the same. The weight of spring, barrel, etc., is taken by the suspension rod G, G and communicated through the lever H to the weight I, which is adjusted so that the lever floats between the stops. The cord, K_1, K_2, K_3, K_4, running over the guide pulleys, L_1, L_2, L_3, communicates the pull due to friction to the Salter's spring balance, where it can be measured accurately. The two pulls on the cords K_1 and K_4, being equal and opposite and at the same distance from the bearing B, do not disturb the equilibrium of the suspended parts. The following *rationale* of the action of the machine will be readily understood by our readers in connection with Fig. 22 :—

If L = total pressure put on journal by spring in lbs.,
r = radius or half diameter of journal,
R = half the distance between the two cords K_1 and K_4,
$2P$ = pull shown on spring balance in lbs.,
f = co-efficient of friction.

The moment due to friction at the surface of the journal is balanced by the two equal pulls on the cords K_1 and K_4, and consequently equal to $R \times 2P$; if we divide this by r, it will give us the frictional resistance in lbs. at the surface. But we have already defined the co-efficient of friction f as being

$$= \frac{\text{Total force of friction or frictional resistance}}{\text{Total press exerted on surfaces in contact}},$$

so that

$$f = \frac{R \times 2P \div r}{L},$$

which is a very easy form of expression for finding the co-efficient of friction.

Boult's Patent "Cylinder Lubricant" Tester.—This machine is designed to test lubricants at the same temperatures at which they are required to work in the cylinders of steam, gas, and hot-air engines. We have already referred to the importance of good cylinder oils, explaining at considerable length the most important points in connection with them. This machine is designed to meet the two great *desiderata* in a "cylinder lubricant" testing machine:—(1) The maintenance of the friction surfaces at any desired temperature; (2) the application of a comparatively small pressure between the surfaces in contact. The machines already described have no provision for the first of these two, and are designed especially to give considerable pressure, up to and over 400 lbs. per square inch, which is most necessary in the case of heavy running shafts, etc., but in the case of cylinder oils is not required. Fig. 23 gives a general view of the machine, and Fig, 24 shows a section through the bearing, giving a very good idea of the interior of the machine. The heating is effected by a Bunsen burner,

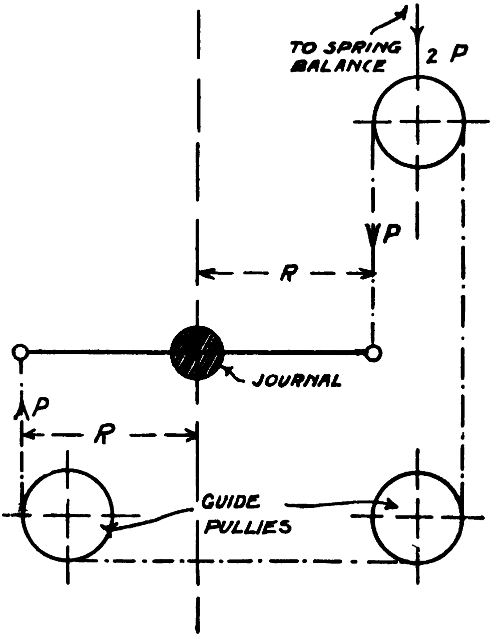

Fig. 22. Section of Prof. Smith's Machine.

F, which heats the oil in the chamber C, C, which runs right round the brass. This chamber is filled with a heavy mineral oil, since this has a very high boiling point; water would never do, as it would boil long before the temperature reached even that which steam has at 30 or 40 lbs. per square inch. A thermometer, A, with its bulb in the heated oil, registers the temperature; tests at the high temperature of 450 deg. Fahr. have been conducted upon this machine.

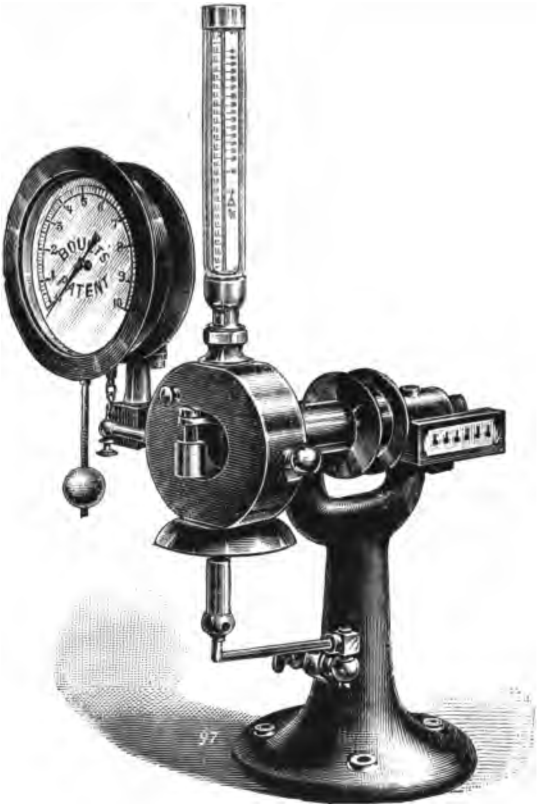

Fig. 23.

The bearing, about 2½ in. diameter, is in this machine in two parts, D, D, with a spiral spring, E, between them to cause pressure between the bearing and the brass. The two screws B, B are for compressing the spring E and allowing the brasses, which are not, as usually, in halves, to be removed from off the journal D, D. The frictional resistance is read off on the dia.

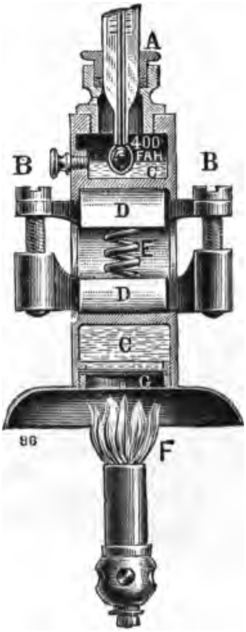

Fig. 24.

shown in Fig. 23, a counter giving the number of revolutions per minute.

The directions for testing the durability of an oil with this machine are briefly:—"Light Bunsen burner, and heat the bath to the temperature corresponding to the pressure the oil will be used under; see the friction surfaces are perfectly clean; these can be examined by tightening the set-screws in order to compress the spring. The cylindrical bath can now be lifted away, when the surfaces are cleaned, a *measured quantity* of the oil to be tested is poured upon them, and the cylinder bath put back in its place. Run the machine until, say, there is a rise of 50 per cent. or 100 per cent. beyond the friction first indicated by the machine. This will show that the virtue of the oil has been practically absorbed, or, in other words, the oil has been tested to destruction. The number of revolutions the machine has made from the start of the experiment till, say, 50 per cent. rise in the friction is noted; this will be the value of the oil as a friction reducer; and supposing a dozen different oils to be tested one after the other, the one that runs longest will be the best of the twelve.

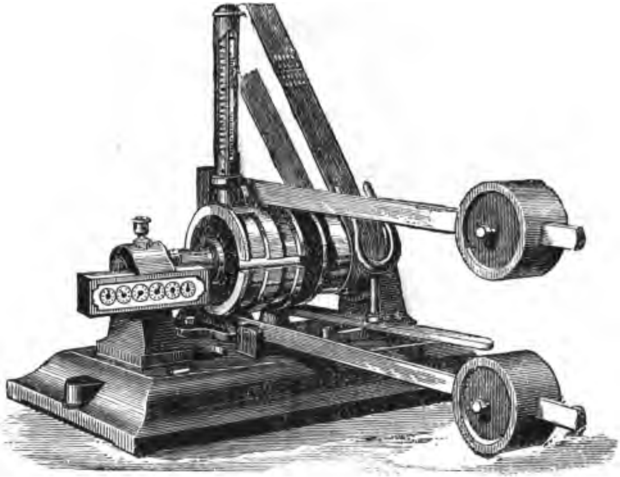

FIG. 25.

Ingram and Stapfer's Patent Oil Tester.—This is one of the first machines ever used for testing lubricants. The principle of its action consists in two brasses pressed upon the bearing by the weights shown. The oil to be tested is introduced in small and measured quantities upon the bearing, and, as in the machine last described, the apparatus is run at about 2,000 revolutions per minute, until the thermometer shows the temperature of the journal to be increased to 200 deg. Fahr. When the lubricant permits the bearing to reach the above temperature, its friction-resisting quality is *worn out;* the oil that permits the greatest number of revolutions is then judged to be the best. In this machine there is no attempt made to measure the actual resistance of friction as in Thurston and Smith's machines, so that we cannot find the co-efficient of friction without some other apparatus added to measure it. The machine is made in two sizes; the first with a spindle 3 in. diameter for general use, the

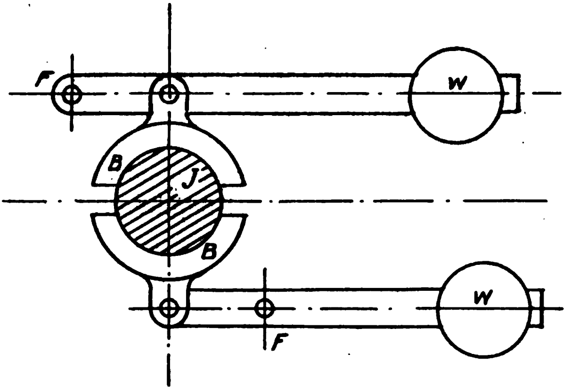

FIG. 26. Section of Ingram and Stapfer's Machine.

second with a 6-in. journal designed especially for railway companies, and for testing thick greases. This machine is also manufactured by Messrs. W. H. Bailey and Co., of Salford; the price of the small machine being £8 10*s.* The illustration (Fig. 26), showing the principle of the machine, we have prepared ourselves; the one showing the general view we are indebted to the manufacturers.

We have now considered, though not altogether at the length that that part of our subject deserves, the means of testing by chemical and mechanical methods oils and other lubricants. We just add, in justice to other inventors of oil-testing machines, that there are many others, as Riehlé's and Ashcroft's, as well as those of Messrs. Tower and Goodman, already described when we were considering friction experiments. These two last were made to meet special requirements, and are not manufactured.

CHAPTER VIII

COMPARATIVE VALUE OF LUBRICANTS

Comparative Value of Lubricants — Principal Classes of Machinery and Oils used.

THE following tables give the relative order in which the prices of the oils—animal, vegetable, and mineral—most generally used for lubrication stand.

It is clear that no oil, no matter how good its origin may have been, will answer for every class of work, so that we feel justified in urging our readers again that they should endeavour to find out the properties required in the lubricant for their several services, and choose accordingly. We have endeavoured to give below, without venturing into wearisome detail, the principal classes of machinery and the oils generally used for such classes, and what our experience shows to be most economical.

Machinery may be divided into the eight following classes.[1] The machines given under each heading as illustrating that class are due to us, as well as the one or more kinds of lubricant suitable for that class, and although one or two extreme cases, as (1), (2), and (8),

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[1] A division due, the writer believes, to Prof. Thurston.

TABLE XVIII.

Oil.	Remarks.	Price per gallon.	
Lard Oil	Finest	4/6	4/9
,, ,,	Ordinary	3/6	3/9
Neatsfoot Oil		3/3	3/6
Tallow Oil		3/6	3/9
Animal Oil		3/-	3/3
Sperm Oil	Finest Southern	4/-	4/3
,, ,,	Intermediate	3/6	3/9
,, ,,	Arctic	3/-	3/3
Whale Oil	Seldom used for lubrication	2/3	2/6
Seal Oil	Chiefly used for burning	2/6	2/9
Cod Liver Oil	Used for leather dressing	—	—
Olive Oil		2/9	3/-
Castor Oil		2/7	2/10
Refined Rape	For lubricating and burning	2/6	2/9
Palm Oil		2/6	2/9
Cheap Natural Mineral Oil		9*d.*	1/-
Best Refined Mineral Oil		1/-	1/6
Cheapest Mineral Oil		9*d.*	1/-

may not be often met with in the practice of many of our readers, yet they are retained to complete the subject.

Classes of Machinery.—1. *Low Temperatures,* such as are met with in machines whose motive power is compressed air, such as rock drills, pneumatic coal cutters, and engines to supply power for small workshops, worked with compressed air from a central station as in Paris and other towns. For this purpose light mineral lubricating oils, especially those of Russian origin, are most suitable, since they are possessed of a very low freezing-point. Price per gallon for such, 1*s.* 6*d.* to 2*s.*

2. *Very great Pressure with Slow Speed.*—Met with under pivots of swing-bridges, where several thousand pounds' pressure per square inch is met with; of course the movement is extremely slow in these cases. Professor Thurston says plumbago or graphite is especially valuable to prevent abrasion under very heavy loads and at low velocities. It should be pure, free from grit, and ground to a flaky powder. A mixture also has been recommended of paraffin and lard oils forced between the surfaces by use of a pressure pump which will carry the heaviest pressures met with under drawbridges.

3. *Heavy Pressure with Slow Speed.*—Under this head come the necks of rolls in rolling mills; cross-head or gudgeon pins, where as much as 1,200 lbs. per square inch is in every-day use, and the shafts of heavy land and marine engines. Semi-fluid oils, as tallow, lard, and castor oil, are suitable. Sperm oil, though often used, is not suitable, since it lacks body; its price, too, is very great. Heavy mineral oils and compounds will, we think, be best, for we have shown in a previous chapter that compound oils are better than either of their ingredients used singly. Mr. Tower's experiments clearly showed that a mineral grease was better than a mineral oil for these cases. Tallow will cost, say, 4½*d.* per lb.; castor oil, 3½*d.*, answering to about 4*s.* 2*d.* and 3*s.* 3*d.* per gallon respectively; sperm oil from 4*s.* to 4*s.* 3*d.* per gallon; compound mineral oil from 2*s.* 1*d.* to 2*s.* 10*d.* per gallon.

4. *Heavy Pressure and High Speed.*—Ordinary steam engine work (not cylinders, for that see 7), but crank pins, bearings, guides, etc., come under this class. They may well be again divided into two classes, heavy and

light, *i. e.* engines ranging from 150 h.p. upwards, and then those below that. For the heavier class neatsfoot and castor oils are used, 4*s.* 3*d.* per gallon and 3*s.* 3*d.* per gallon respectively. For the lighter class lard and tallow oil, as well as olive oil, at, say, 3*s.* to 4*s.* 3*d.* per gallon. Lard oil is generally used on torpedo-boat trials where the limits of pressure combined with high speed are closely touched, but since olive oil is now a little cheaper, that has been used lately. Heavy pale compound oils for the heavier class can be obtained at from 1*s.* 9*d.* to 2*s.* 6*d.* or 2*s.* 10*d.* per gallon, while a lighter kind, suitable for the other class, at from 1*s.* 5*d.* upwards.

5. *Light Pressure and High Speed.*—Say up to 100 lbs. per square inch and above 150 ft. per minute, met with in dynamo machinery, very fast engines for driving dynamos, looms, lace-making machinery, cotton-mill machinery, and so forth. For this sperm oil is much used, although the light natural mineral oils sometimes excel it at prices from 1*s.* 6*d.* to 2*s.* 7*d.*

6. *Ordinary Machinery.*—Under this comes a vast collection of every-day machinery, such as shafting, lathes, planing and shaping machines, drilling machines, pumps, cranes, circular and band saws, planing and other wood-working machinery, mortar mills, mowing, reaping, and threshing machines, traction engines, tramcars, flour milling machinery, the lighter class of shearing, punching, and riveting machinery. For these machines, although lard oil, sperm oil, olive, neatsfoot or tallow oils are suitable and very efficient lubricants, yet we consider their price, ranging as it does from 2*s.* 11*d.* to 4*s.* 3*d.* per gallon (see Table XVIII.), leaves the door wide open for a cheaper

substitute in the form of natural or compound mineral oils, at, say, from 1*s.* 6*d.* to 2*s.* per gallon, some, in fact, as low as 1*s.* A very good engineers' shop oil can be had for about 1*s.* 10*d.* per gallon, which will answer for shafting, machines, and all purposes save cylinder lubrication, and be found the most economical, upon the whole. If the waste that takes place by machine hands, fitters, etc., could be avoided, an oil costing 2*s.* 6*d.* per gallon would be found perhaps somewhat better; but it is almost impossible to prevent waste, so that it is best to use the cheaper oil. If circumstances are such that great care can be exercised in the use of oil, we should say use one about 2*s.* 6*d.* per gallon, on account of its greater endurance. As to lard oil, in warm weather we do not consider there is any oil to equal it weight for weight; but being very expensive we do not consider it economical to use. Men will waste oil, nor will they waste less, generally speaking, if a good one is put into their hands, so that it is better to waste the same quantity of a cheap oil than the same quantity of a dear one. In cold weather lard oil becomes thick and is a little troublesome to use. For shafting in general use—shop shafting we refer to, not heavy marine or land shafts—loose pulleys and such like, we consider the use of solid grease lubricant is advantageous. A thoughtful and experienced engineer of our acquaintance, who has the entire management of the works of a well-known country firm, has kindly furnished the writer with the following particulars in connection with the use of solid grease for shop shafting, having used it for over ten years: "Some manufacturers claim that there is a saving of 90 per cent. in using grease in lieu of oil.

We are quite sure that there is a large saving, but not to the extent claimed, when oil is properly used and the lubricators kept in proper working order. We have been using grease lubricant for several (10 to 12) years, and at the present time have seventy-nine bearings running with same, and are still extending its use, both on account of its economy and great cleanliness. As to cost, 1 lb. of grease, costing 68*s.* per cwt. (7¼*d.* per lb.), will last one month for our seventy-nine bearings, and they would require in the same time 1¼ gallons of mineral oil, at 1*s.* 10*d.* per gallon (2*s.* 3½*d.*). Our bearings carry ordinary shop shafting, and vary from 3½ in. to 1¾ in. in diameter, working on an average sixty hours per week, and running at eighty revolutions per minute. We consider that the grease and oil are equal in value as lubricants when the foregoing quantities are used, both keeping the bearings in good order. Grease can be bought as low as 38*s.* per cwt., but we always use the best at 68*s.*, and have no experience with the other qualities. We have not noticed the least difference in the grease that we use due either to summer heat or winter cold."

Complaints are often made that in hot weather the solid grease becomes fluid and runs away like water, and we are glad to give the foregoing as rebutting evidence. It is not a very great assumption, we venture to say, that such action in hot weather was due to using a cheap and bad grease. It may be of interest to some of our readers to notice that the lineal velocity of bearing surfaces varied from 40 to 80 ft. per minute in the above case, and that the bearings in the case of solid grease on an average consumed each 2½ ozs. of lubricant per annum, and the case of oil 26 ozs. per annum. This last gives, at, say, 310 working days per annum and 480 drops in a fluid ounce, about four drops per hour. This appears very small, but one-tenth the amount is all that is used in the case of solid grease, or four drops per day, reckoning ten hours per diem. The difference no doubt can be accounted for in great measure by the fact that so much is wasted by the escape of fluid oil at the ends of the journal. It is to be noted that in the above instance a saving of 75 per cent. on actual cost has been effected by the change. We shall later on, when describing *the best means of lubricating,* illustrate two or three very widely-used forms of lubricators for this material.

7. *Steam Cylinders.*—In touching this most important section we shall ask the permission of our readers to recall their attention to what we said in Chap. V. in connection with Acidity and Decomposition, not, however, to go over all the ground again. It was there pointed out that tallow consists of fatty acids—stearic and oleic—and glycerine, which acids are liberated by the action of water and high-pressure steam. It will be seen that the cylinder of a steam engine offers these conditions, and the fatty oils—animal and vegetable—become sooner or later reduced to their simpler elements. Mineral oils are entirely free from this objection; and when they have been prepared for use by steam heat only, without the use of sulphuric or other mineral acids, they are most suitable for use in cylinders. There is another evil, too, attendant on the use of tallow in connection with the boiler. Mr. J. F. L. Crosland, in a discussion at a meeting of the Association of Employers, Foremen, and Draughtsmen of the Mechanical Trades of Great Britain, said that

"all boiler engineers were aware that where tallow was used in the cylinders of a non-condensing engine which delivered exhaust steam into the feed-water cistern, and where the water contained carbonate of lime (chalk), they were almost sure to have the burning of furnace plates. It was a difficulty which did not happen where the water was pure—as in Manchester or Glasgow. Wherever the water was quite pure and free from deposit there was no trouble. . . . Where the water did not produce that particular kind of deposit the grease did not do any harm—that was to say, tallow, or suet, or any animal or vegetable oil, seemed to be innocuous"—that is, as far as the *boiler* is concerned,—"but wherever the water produced a fine, floury deposit, such as was found in the south of England, boiler flues were liable to collapse. He remembered that at the Scarborough waterworks the collapses of flues were so frequent that it could only be concluded that they were purposely caused, and inspectors were sent day and night to watch the boilers, and it was found that, with eight or nine inches of water in the glass, the furnaces went down before their eyes. The deposit in this instance was of the floury nature described, and very little was sufficient to cause collapse as long as any grease was allowed to enter the boiler. All imaginable devices were made to strengthen the flues of these boilers, but nothing would keep the furnaces in order till tallow was prevented from entering. *It was a noteworthy thing that, where cylinder oil was used with the same water, it did not seem to have this objectionable result.*" This testimony to the value of mineral oils is very decided, and has the advantage of being entirely unbiased. If, as is sometimes necessary, it is advisable to add a small percentage of animal or vegetable oil to a mineral oil to impart more *body* to it, the above-mentioned injurious defects are reduced in a very great degree when compared with the use of pure vegetable or animal oil.

Surface Condensing Marine Engine Cylinders.—In this class of cylinders a pure hydro-carbon mineral oil is most essential, since the feed water is used over and over again. The higher the pressure, and the larger the engine, the heavier the oil should be. The price per gallon ranges between 2*s.* and 3*s.* 9*d.*

Heavy Horizontal Land Engine Cylinders.—In these engines—say from 300 h.p. upwards—the lubricant is subjected to a much higher pressure, due to the weight of the piston bearing upon the cylinder, and, consequently, a thicker oil is advisable in such cases than is used in vertical engines. Added to this, land engines are usually jet-condensing—if surface-condensing they would come under the previous class—and, therefore, the admixture of a little animal or vegetable oil to give body is admissible. The prices would not vary much from the previous class.

Ordinary Land Steam Engine Cylinders running from small shop engines, traction engines, etc., up to about 300 h.p. For such we should recommend a pure, or nearly pure, hydro-carbon oil. These will run from 1*s.* 10*d.* to 3*s.* 9*d.*; we recommend one at about 3*s.*, and believe that such will be found to give every satisfaction.

Small Engines, say up to about 6 n.h.p., working with 30, 40, or 50 lbs. of steam, will thrive under the class of oil we indicated for general shop use at 1*s.* 10*d.* per gallon, since they are seldom or never fitted with sight-feed lubricators, which are necessary for the economical use of higher-priced lubricants.

Gas Engine Cylinders.—For these, owing to the very high temperatures occasioned by the explosion of the gas, it is necessary that oils of the highest quality, as regards flash-point and degree of refining, should be employed. Pure hydro-carbon oils may be used, but mixtures of mineral and fatty oils, if carefully prepared and their proportions suitably adjusted, possess certain advantages over the unmixed mineral oils. A light amber-coloured oil is to be generally preferred to a dark one; such can be bought for 2*s.* 4*d.* per gallon; one at 3*s.* per gallon should be the most expensive allowed.

Hot Air Engine Cylinders come under the same conditions as ordinary land steam engine cylinders, and the remarks made above will apply to this case.

8. *Watches and other Delicate Mechanism.*—Watches, clocks, mathematical and nautical instruments, steam engine indicators, sewing machines, and such kind of machinery, need very refined oils, and, consequently, fancy figures are charged for them. All the finer oils are *got up* for these purposes—sperm, olive, porpoise, bone, and almond oils, as well as light mineral lubricating oils, are sold either pure or as mixtures, sometimes under fancy names and in small quantities, for which the consumer has the inestimable privilege of paying 300 or 400 per cent. over their actual value. The small amount used, too, has, without doubt, a tendency to raise the price. It should be mentioned that in all the foregoing cases the prices mentioned are for large quantities; where a few gallons only at a time are bought the figures are sure to be a little higher.

Stauffer's Lubricant is one of those compositions known as "solid oils," which find, and we consider justly, favour under certain conditions of work—the absence of drippings and consequent cleanliness of bearings and economy in lubricant being undeniable advantages. To say, however, that it is suitable under every condition is going too far, and our own opinion is that a properly designed oil-bath bearing would be found to work with less friction and quite as economically, but, unfortunately, such bearings are not easy of application and would require, generally, entirely new bearings, which would mean as many pounds in outlay as a lubricator to use solid grease would shillings.

Stauffer's Lubricant is guaranteed to be free from acid and also not to gum. Its lowest melting-point is 180 deg. Fahrenheit and 212 deg. Fahrenheit, according to its grade. It is made in five grades :—

A1 grade	68*s*. per cwt.
A „	56*s*. „
B „	48*s*. „
Centi-Grade	48*s*. „
C „	38*s*. „

Of these we recommend the best at 68*s*. per cwt., unless for very rough work. The Centi-Grade is made especially for high temperatures and hot climates; its melting-point is 212 deg. Fahrenheit. If had in 3½ cwt. casks a reduction of about 4 or 5 per cent. is made from the above prices.

Stauffer's Lubricators.—These are made in several forms to suit different purposes, but the principle of their action is always the same. Fig. 27 illustrates the ordinary type, and the table on p. 112 gives the sizes

TABLE XIX.

SIZES AND PRICES OF STAUFFER'S LUBRICATORS.

MESSRS. TRIER BROS., MANUFACTURERS.

No.	0	1	2	3	4	5	6	7	8	9	10
	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.	In.
Outside Diameter of Grease Cup ... ... A	$\frac{1}{2}$	$\frac{3}{4}$	1	$1\frac{3}{8}$	$1\frac{3}{4}$	$2\frac{1}{8}$	$2\frac{5}{8}$	$3\frac{1}{8}$	$3\frac{5}{8}$	$4\frac{1}{4}$	$5\frac{1}{8}$
Depth of Cup C	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{15}{16}$	$1\frac{1}{16}$	$1\frac{3}{16}$	$1\frac{5}{16}$	$1\frac{7}{16}$	$1\frac{9}{16}$	2
Length of Shank ... ... B	$\frac{5}{8}$	$\frac{3}{4}$	1	$1\frac{1}{16}$	$1\frac{1}{8}$	$1\frac{3}{16}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{7}{16}$	$1\frac{9}{16}$	$2\frac{3}{8}$
Diameter of Shank, Size of *Gas Thread* ... D	—	—	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{1}{2}$
" " " Plain, without Thread ...	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{4}$
Approximate Diameter of Shafting for which Lubricator is suitable	$\frac{3}{8}$	$\frac{5}{8}$	$\frac{7}{8}$	$1\frac{1}{4}$	$1\frac{3}{4}$	$2\frac{1}{2}$	3	4	5	6	8
Price—*All Gun-metal*, Polished, as in Fig. 27 ... ... Each	9*d.*	1/-	1/2	1/6	2/-	2/4	3/-	4/-	5/6	7/-	9/6

corresponding to the letters on Fig. 27 for the ten different sizes in which they are made. We have specially drawn this figure and tabulated the dimensions in order to enable our readers, before ordering, to know what space they will occupy. The prices given are for all gun-metal lubricators, but they can be had in special toughened fluted iron at from 30 to 40 per cent.

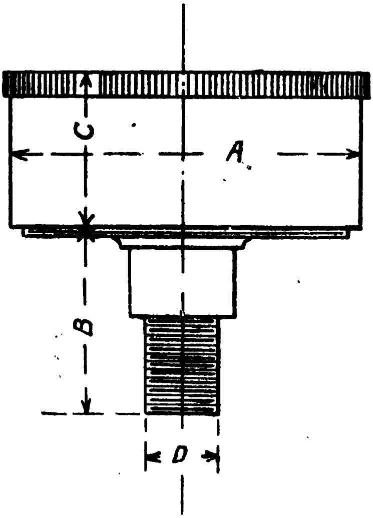

Fig. 27. Stauffer's Lubricator.

cheaper; there is also a slight reduction on the dozen. Since, both with the iron and gun-metal cups, the shanks have been found to break off, Messrs. Trier Bros. have introduced a New Patent Lubricator, in which a wrought-iron nipple is screwed into a toughened cast-iron shank; when these shanks are fitted with polished gun-metal caps they are recommended as superior to all gun-metal lubricators, being considerably cheaper yet fully equal to them in appearance.

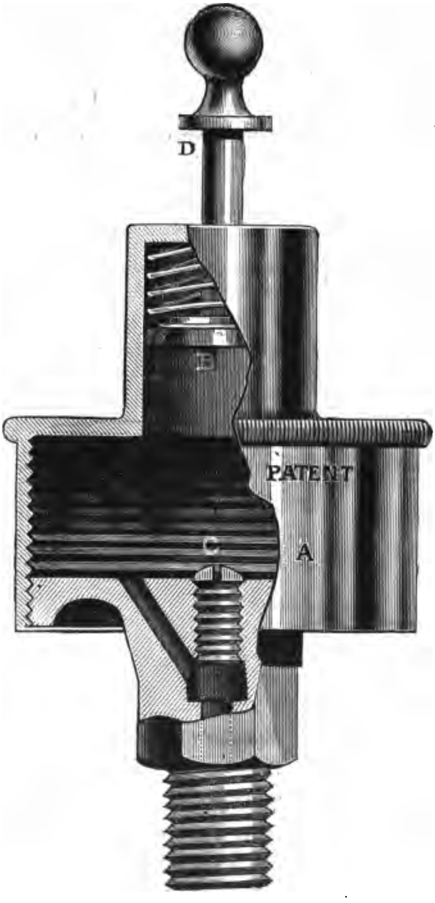

FIG. 28.

Patent Tell-Tale Automatic Lubricator.—Fig. 28 shows this form of lubricator; it has a projecting cylinder on top of the cap A, of much smaller diameter than the cap, in which a cup leather piston, B, is pressed down by means of a weak spiral spring. The piston rod D, passing through the top of the cylinder, serves not only as a guide for steadying the piston but also as a *tell-tale*, showing when the cap wants another turn down, which again forces the piston B up, bringing the spring again into action. The screw C is to regulate the supply to the bearing.

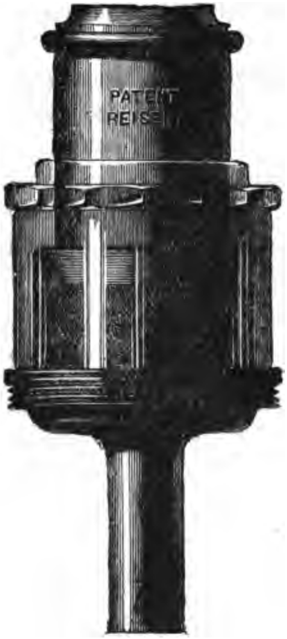

FIG. 29.

Reisert's Patent Self-Acting Lubricator.—Fig. 29 is an outside view of this apparatus. In all the above-described cups the cap needs to be given a turn down by hand every week or eight days, but this is entirely self-acting. A soft cup leather piston works in a strong glass reservoir. This piston has attached above it a hollow stem, which is loaded with shot or washers just sufficient to keep up a gradual and constant supply of grease. The four following advantages are claimed for it:—

(1) *Contents always visible,* since reservoir is of glass.

(2) Lubricant cannot squeeze past piston nor can piston jam, giving *certainty of action.*

(3) Pressure on piston being constant, supply of grease is regular, ensuring *better economy.*

(4) Since reservoir can easily be detached from base *it can be easily filled.*

Prices, each, are as follows:—

Interior diameter and depth of glass ...	1⅛ in.,	1½ in.,	2 in.,	2⅜ in.,	2¾ in.
Price with screwed shank,	1*s.* 9*d.*,	2*s.* 3*d.*,	2*s.* 9*d.*,	3*s.* 6*d.*,	4*s.* 3*d.*

CHAPTER IX

DISTRIBUTION OF LUBRICANTS

The Distribution of Lubricants—Cylinder Lubrication—External Engine Lubrication — Lubrication of Shafts — Magnolia Anti-Friction Metal.

In considering the question of lubrication in actual every-day practice it must be borne in mind—

(1) That sperm is the dearest and mineral the cheapest oil.

(2) That sperm oil has the least weight-carrying power, though the lowest co-efficient of friction.

(3) That mineral oil and mineral grease carry very high loads.

(4) That for high pressures a thick lubricant is better than a thin one.

(5) That for low pressures, combined with high speeds, the best lubricants are best refined light petroleum and best southern winter bleached sperm.

(6) That compound oils have advantage over either pure vegetable or mineral oils, save in the case of cylinder lubrication.

(7) That as little oil as possible should be used consistent with good results, and as thin a one as the weight to be carried by journal will permit without squeezing out.

(8) That relatively more power is expended in driving, *i. e.* overcoming friction, in light machinery than in

heavy machinery, and therefore users of such should expend more care in looking to value of lubricants, etc.

(9) That an account of the various lubricants used

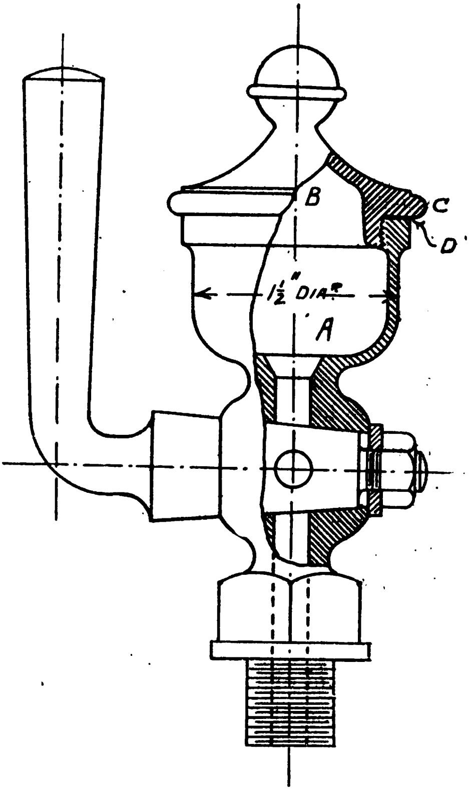

Fig. 30.

should be kept with as much detail as possible, and also that tests should be made especially with regard to viscosity, this test being second only to best testing machine methods.

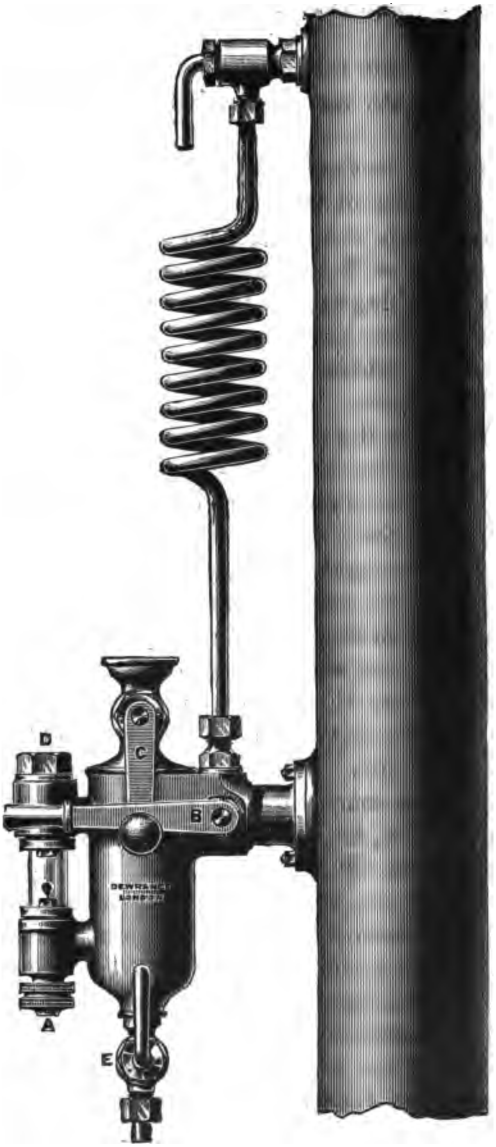

FIG. 31.

(10) That the lubricant should be chosen, not haphazard, but with regard to the kind of work it has to perform.

(11) That continuous lubrication, such as is provided by means of an oil bath, is the very best; but that in practice this is not always, nor even frequently, convenient, and so in adopting another method we must aim at something as near to it as possible.

(12) That our aim, if possible, should be to eliminate any metallic friction in journals, *i. e.* they should be entirely oil borne.

(13) That a dry place on a journal is the beginning of all trouble, therefore give plenty of grooves.

(14) That some journals run best a little hot.

(15) That the oil tubes should be frequently taken down and thoroughly cleaned inside.

(16) That the total amount of friction on a journal is decreased by cutting away at sides.

(17) That the wrong place to introduce lubricant is the place where the greatest pressure is, and the best place is at the side which is leading on to the place of greatest pressure. Where there is reciprocation, as in the end of a connecting rod, this is not such an important matter, for time is then given to permit the oil to get between the surfaces.

(18) That for a very high velocity the bearing should be long, the ratio of length to diameter being between 3 and 4, and for slow speeds with heavy pressures a short bearing is best, the length being from 1 to 1½ times the diameter of shaft or pin.

(19) That the amount of pressure per square inch allowed in usual practice is as follows:—

Main Bearings of an Engine should never exceed	600	lbs. per. sq. in.
Average	300	,,
Crank Pins never exceed	500	,,
Average	400	,,
However, in Paddle Engines which move slowly as high as is used	800 or 900	,,
Crosshead and Gudgeon Pins, *i.e.* small end of connecting rod should never exceed	1,200	,,
(Very small angular movement, and very slow.)		
Guide Blocks never exceed	100	,,
Average	40	,,
Ordinary Shafting never exceed	300	,,
Average	100	,,

(20) That proper lubrication does not consist in pouring a lot of oil into a bearing and chancing where it goes to, but in maintaining a film of oil between shaft and bearing.

(21) Pay great attention to the distribution of the lubricant and use, whenever possible, automatic means.

This last consideration introduces us to the subject of this chapter, and it is with no hesitation we say that it is *the* most important factor in lubrication. This will be best treated if we consider, first, cylinder lubrication; second, external engine lubrication; and last, shaft and machine lubrication.

Cylinder Lubrication.—The most common form of lubricator for this purpose is something similar to Fig. 30. This is fitted either upon the top cover of cylinder or upon the valve chest, in some cases in both places. The figure gives a view of a very inferior class of cup used only for small cylinders, say up to 7 in. or 8 in. diameter, as are frequently met with in cranes and such like. A much better form is with another cock and handle introduced above the cup A in place of the screw-cap B, which is difficult to keep tight under steam. There are two objections to this form: (1) that the supply is at very long intervals, and (2) that it is not automatic. Two other designs fulfil these conditions, namely, the displacement cup and the sight-feed lubricator, the latter possessing the advantage of regulation, and also that the actual number of drops fed per minute can be seen. The displacement cup is due, if we remember aright, to Ramsbottom, and depends upon the condensation of a small portion of the steam in the cylinder forming a drop of water, which sinks through the oil in the cup and displaces an equal amount of oil, which finds its way into the cylinder. This is in use to a considerable extent on locomotives, but it is not generally used elsewhere. We show in Fig. 31 a sight-feed lubricator made by Messrs. Dewrance and Co., London. The illustration plainly shows the way the attachment is made to the steam pipe. The upper cock admits steam into the coil condenser, from which it passes down into the body of the lubricator and gradually displaces the oil, which passes in drops up the glass tube into the steam-pipe by the lower attachment. The oil is introduced into the lubricator by first closing the handle B, which liberates the handle C and allows the cock to be opened between the cup and body of machine. The cock E is to drain off the condensed water from the lubricator.

The Crosby "double connection" pattern sight-feed lubricator (see Fig. 32) has two connections with the steam pipe, one being made with the condenser, and one with the sight feed. The pressure of steam in both is the same, but the steam coming in contact with the cool surfaces of the condenser condenses and fills it and the pipe leading to the oil reservoir with water; then the hydrostatic head due to this column of water forces the oil out of the lubricator through the sight glass into the oil delivery pipe, and thence into the steam pipe. As fast as the oil is fed out, its place is taken by water from the condenser, until the oil reservoir becomes entirely filled with water. The oil forced out of the reservoir passes through the feed valve F, and through a small tube or nipple at the bottom of the sight glass; it can then be seen, as it passes to the steam pipe, rising drop by drop through the water with which the glass is filled. When the oil reaches the steam pipe it becomes mixed with the steam, and is carried to all the internal parts of the engine. The feed can be regulated by the valve F so that just the right quantity will be fed in continuously to keep all the rubbing surfaces constantly covered with a film of oil and consequently perfectly lubricated. It is evident that this result will

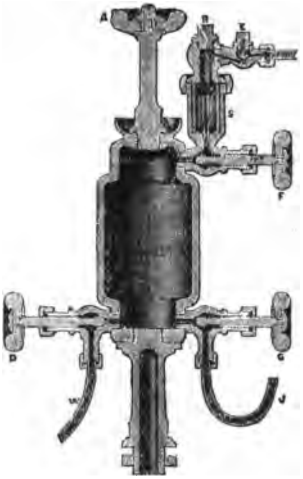

FIG. 32.

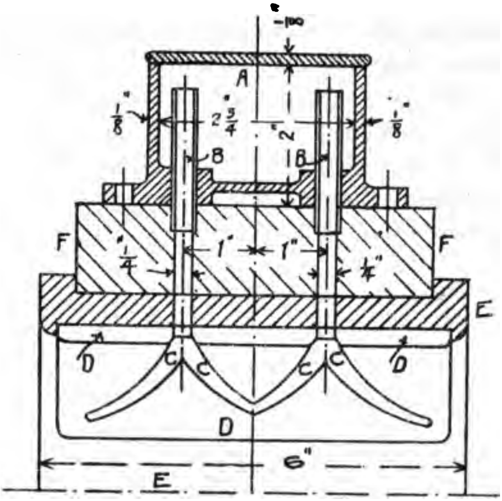
.100
A
1/8"
2 3/4"
2"
1/8"
B
B
F
F
1/4"
1"
1"
1/4"
E
D
C
C
C
C
D
D
6"
E

FIG. 33.

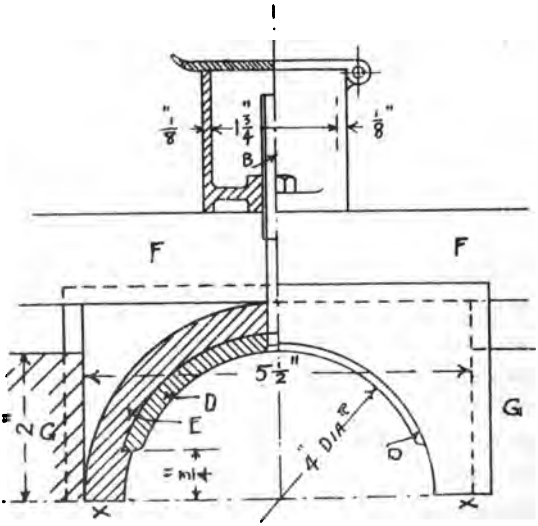
1/8"
1 3/4"
1/8"
B
F
F
5 1/2"
D
E
G
2"
G
4" DIA
D

FIG. 34.

be attained with a minimum expenditure of oil. This style of lubricator can be fixed in any convenient place,

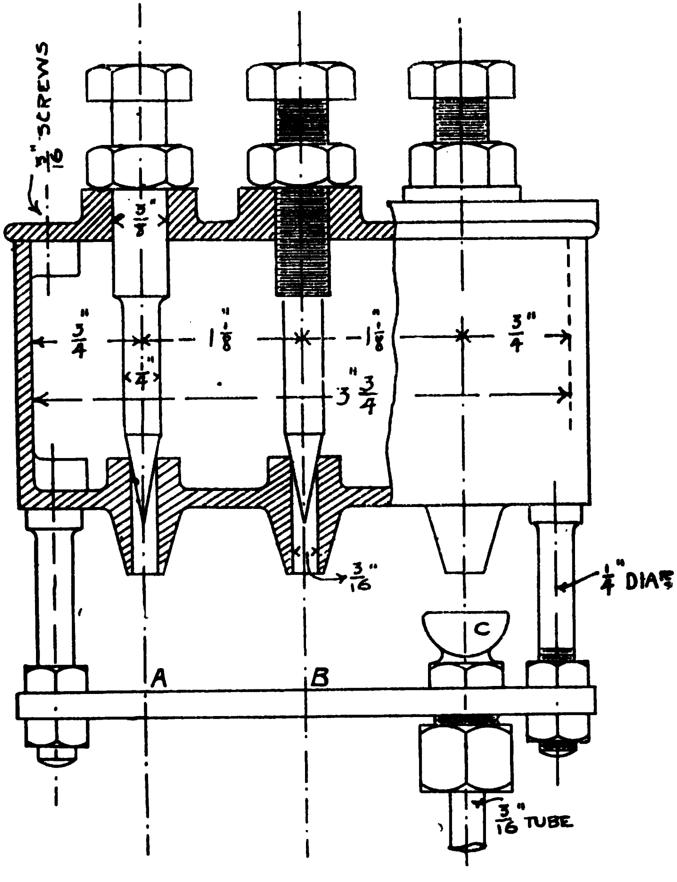

FIG. 35.

the oil being conveyed by means of copper pipes to the cylinder or cylinders.

External Engine Lubrication.—Main bearings, slides, connecting-rod ends, eccentrics, and eccentric rod ends.

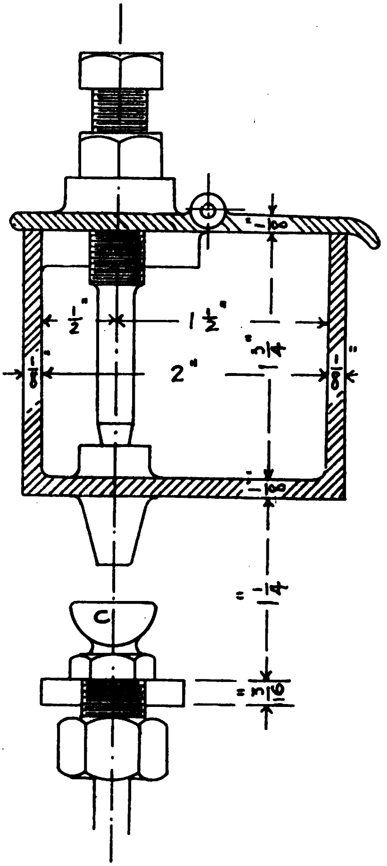

FIG. 36.

Taking the first, main bearings, we have drawn Figs. 33 and 34 to show a thoroughly good method of oiling this class of bearing. The example is taken from actual work, and is in use on the bearings (4 in. diameter, 6 in. long) of a fast-running marine engine (250 revolutions), indicating over 200 h.p. A is a light brass box, with two brass tubes (B) brazed in to form the syphons; it is secured to the cap (F, F) by means of two set screws; the brass (E, E) is of gun-metal, into which is run white metal to form the shaft-bearing. It will be observed that the white metal is not placed at the sides of bearings for a distance of ¾ in. each side of the centre line, ½ in. in all each side. C, C, C, C are oil channels cut into the white metal; it is most necessary for good working that these should be ample. Figs. 35 and 36 show an oil box for lubricating the connecting rod, piston rod, and guides of the above engine, one such box being fitted upon the lagging of both high- and low-pressure cylinders. This possesses two valuable features: (1) The feed can be adjusted to a nicety, and (2) then the drops can always be seen dropping from the box into the cups, which are fitted with copper pipes (A, B, and C), and lead where desirable. No means of attachment have been shown, but such will readily suggest themselves according to the situation. The removable bar holding up the copper pipes (A, B, and C) allows of suggestion No. 15 being carried out without trouble.

Turning now to connecting-rod lubrication, various ways suggest themselves, according to the size of bearings, type of engine, and its velocity. Fig. 37 shows a very small syphon lubricator, A, suitable for use on bearings about 1¼ in. diameter by 2¼ in. long, also for eccentric rods. It is screwed into the cap C of the bearing B. D is a small brass tube inserted to make a syphon. There is, however, one great objection to it, and that is the *open top* permitting dust to collect; but for cheap and to some extent rough work, like is met with in cranes, they answer every purpose and are extremely cheap. Fig. 38 shows a needle lubricator suitable for the same service on a horizontal slow-running engine, say up to 60 revolutions per minute. A is a

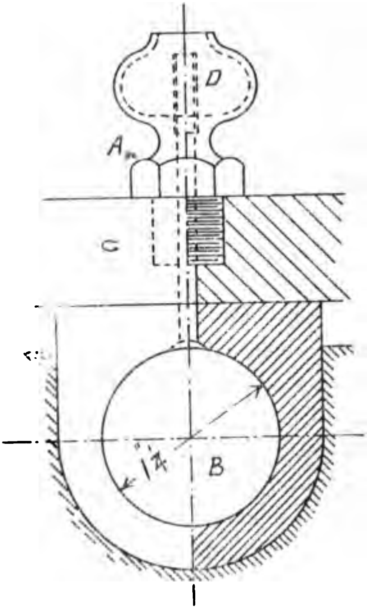

Fig. 37.

brass casting formed to receive the glass cup B, which is held in position by the brass cap C. D is a fly-nut with a leather washer E underneath, and is used for filling lubricator; F is the needle made to bear upon the journal. For much faster running horizontal engines, a lubricator is generally designed which depends upon centrifugal force in its action, but our space forbids a description. Again, in very fast-running engines, like

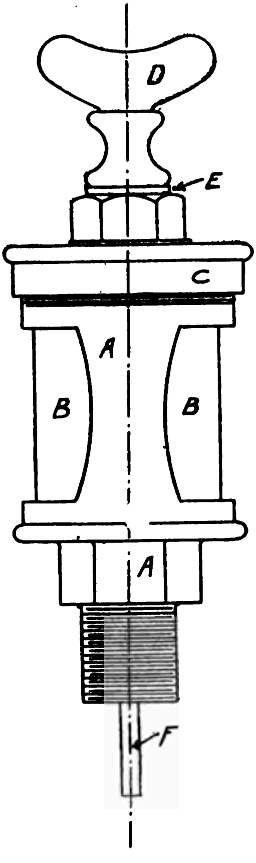

Fig. 38.

Willan's Central Valve or Globe Compound, the connecting-rod ends run in a bath of oil, by far the best method, but of course inapplicable to most engines. In

Figs. 39 and 40 we give an example of an oil cup for an inverted type marine engine. A is a brass cup (shown in section in accompanying view) which is fastened to the top end of connecting-rod B, the lubricant passing down the copper pipe C to the big end of the rod. In Fig. 40, which shows the interior of cup,

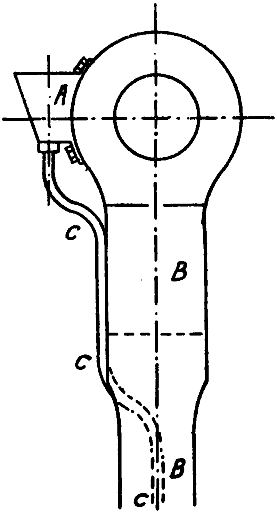

Fig. 39.

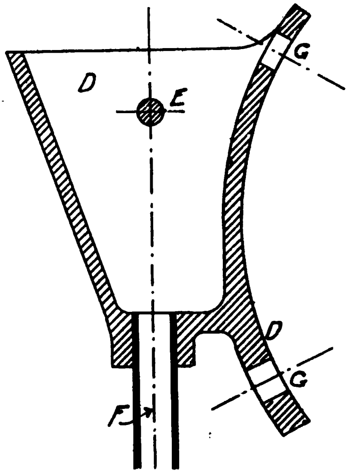

Fig. 40.

G, G are the studs to secure it to rod and F the copper tube. One of the pipes shown in Fig. 35 is so led that it will drop into the top and thence to brass. The crossbar E is to rest the end of oil-can spout upon in the event of oiling by hand. The crosshead pin is lubricated in the same way by a similar cup fastened to the crosshead. The cup is about 1 in. wide, and is suitable for a 4 in. by 6 in. bearing.

We show in Fig. 41 a neat method of lubricating the piston rod of an horizontal engine, 1¼ in. diameter —we have shown it thus merely for the sake of space, but it can be used for rods up to 3 in. diameter; it is arranged on the syphon principle. It necessitates the flange of the gland being made slightly thicker than is usual. For larger rods 4 in. to 6 in. diameter an oil cup is often fixed which drops the oil upon the rod just outside of the gland.

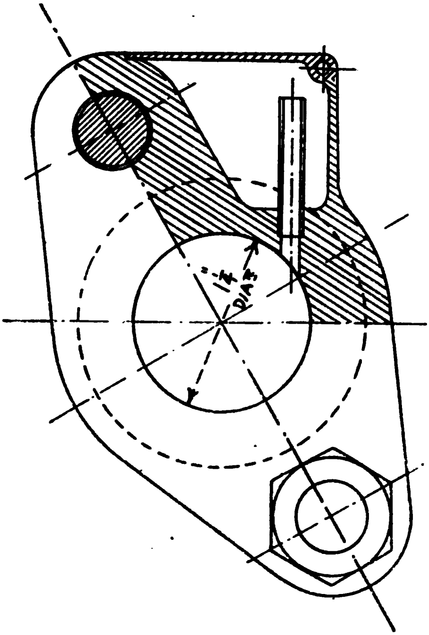

FIG. 41.

Lubrication of Shafts.—These may be oiled by the following methods:—Solid grease cup, syphon grease cup, needle lubricator, pad, or oil bath. The solid grease cups we dealt with in the last chapter, and merely say again that it is a much-used method. The

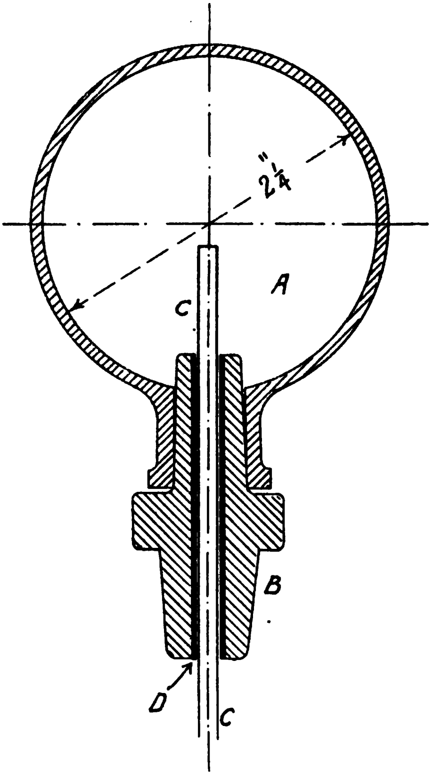

Fig. 42.

syphon cup may be as simple as that shown in Fig. 37, as good as that shown in Fig. 33, or as elaborate as fancy or prejudice may dictate. We give another illustration, Fig. 42, showing the simplest form of the needle lubricator. Fig. 38 also gives another which, if neater and a little more expensive, is not by any means more effective. Figs. 43 and 44 give an approximation to an oil bath. This bearing is made by the Globe Engineering Company, and is lined with magnolia

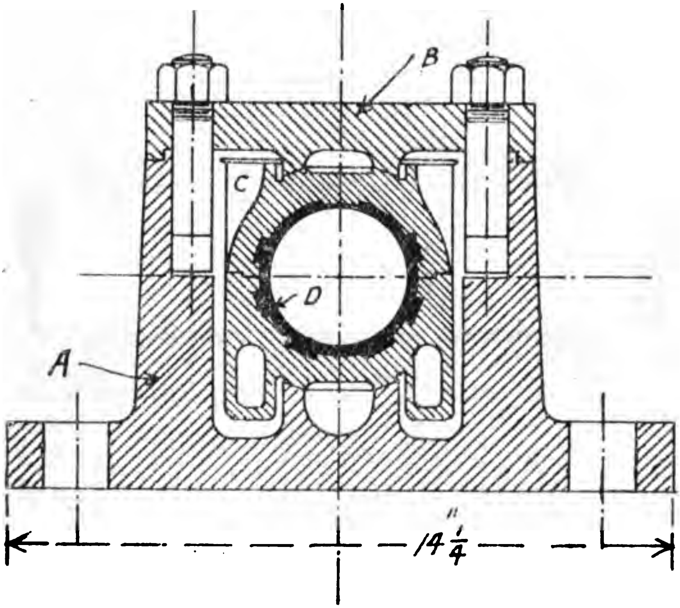

FIG. 43.

Looking first at the longitudinal section (Fig. 43), there is a main casting A with a cap B, turned inside to a portion of a sphere allowing the casting C, C to move in it. This casting is bored out and lined with magnolia metal at D, D, D; in the spaces of G, G two chains revolve which pick up the oil in the bottom of bearing and convey to the top of shaft, the spaces F, F, F being cut out on purpose to allow of the distribution of the oil. In the transverse section in Fig. 44 the method of securing the magnolia to the casting by dovetail grooves is clearly indicated. The makers say that one charge of oil will suffice for a twelvemonth in this type of bearing.

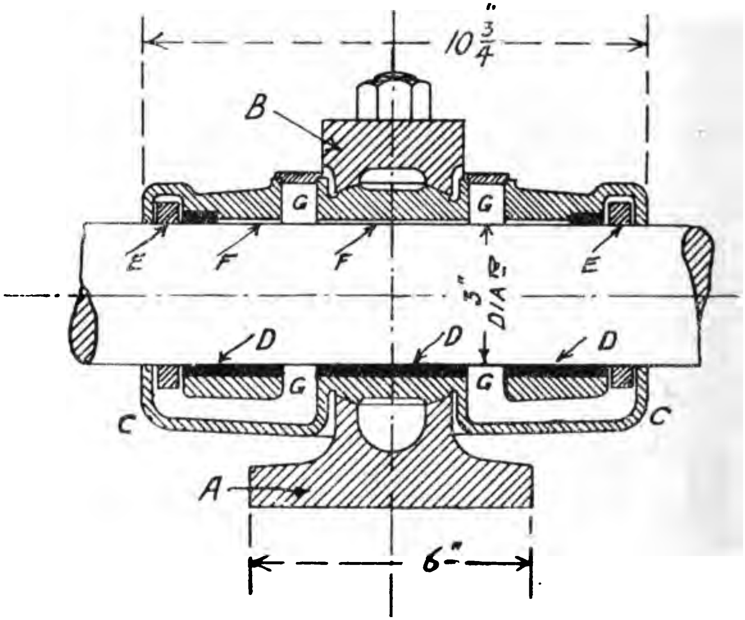

FIG. 44.

Magnolia Anti-friction Metal.—Among the many anti-friction metals in every-day use we must refer to the "Magnolia" metal. Several important advantages are claimed for it, not the least of which seems to be its adaptability for both light and heavy work. It is equally suitable for the heaviest plate-mill roll bearings and the lightest spindles of a spinning frame. Many tests have been made upon it both in this country and the United States. Another good feature is its capability of running for a considerable period of time without lubrication. This will be of interest to our readers, since if from carelessness or accident the supply of oil gets cut off, a bearing lined with magnolia may be trusted to run cool for a considerable time without injury to the shaft.

Another adaptation is also worthy of notice, namely, the use of magnolia with perfect success for the faces of marine and locomotive slide valves, where lubrication is anything but perfect, and where, we are informed, it can be adopted for steam pressures as high as 225 lbs. per square inch. The slide valve is, of all the moving parts of an engine, the place where the co-efficient of friction is the highest, 0·2 being generally used in making calculations for the stresses set up in the rods, etc.

The way in which a valve face is usually lined up is by the insertion of dowels of the metal, the holes being undercut, and after the metal is poured in and cooled, it is well hammered into place. Another point about it is the fact that it can be re-melted time after time without deterioration through oxidation.

We may, lastly, turn to some special tests of the metal in conjunction with others, remarking, by the way, that with special machines, tests as to pressure and velocity of rubbing surfaces can, of course, be carried out considerably beyond the limits which necessarily obtain in actual work.

The Government of the United States instituted some experiments in order to arrive at the comparative values of the magnolia and Parson's White Brass No. 2. The results are published in the Report of the Secretary of the Navy, U.S.A., 1888. It would be occupying too much space to reproduce the whole report, but it concludes thus:—"A summation of the results of the experiments on this machine shows the great superiority of the magnolia metal over the Parson's White Brass. The magnolia, even after the heaviest stress, retains a smooth polished surface. This metal has also been tested in the friction machine up to 600 lbs. per square inch—the limit of the machine." We understand that it is in use in over twenty of the United States war vessels.

Another test is that carried out by an influential firm in the North of England. The conditions were as follows:—A shaft 3 in. diameter was run in two bearings 4½ in. long each and 16¼ in. centres; between these was placed the magnolia bearing to be tested. This was 3 in. by $3\frac{3}{16}$ in., giving an area of 9·56 square inches. The total load upon the bearing was 3,024 lbs., giving thus an average pressure of nearly 500 lbs. per square inch on the bearing. The revolutions were 183 per minute, giving a lineal velocity at surface of bearing 145 feet per minute. The run was for *over four hours*, and *oil was supplied for the first ten minutes and then completely shut off.*

MAGNOLIA BEARING.

Time.	Temperature. Fahr.	Time.	Temperature. Fahr.	Time.	Temperature Fahr.
3.50 P.M.	71·6	4.15 P.M.	88·7	4.40 P.M.	93·2
3.55	74·3	4.20	89·6	4.45	91·4
4.0	77·0	4.25	91·4	4.50	91·4
4.5	80·6	4.30	93·2	4.55	91·4
4.10	86·0	4.35	93·2	5.0	91·4

Time.	Temperature. Fahr.	Time.	Temperature. Fahr.	Time.	Temperature. Fahr.
5.5 P.M.	93.2	6.5 P.M.	145·4	7.5 P.M.	136·4
5.10	93·2	6.10	138·2	7.10	136·4
5.15	93·2	6.15	138·2	7.15	139·1
5.20	95·0	6.20	138·2	7.20	140·0
5.25	102·2	6.25	138·2	7.25	138·2
5.30	104·0	6.30	138·2	7.30	138·2
5.35	111·2	6.35	137·3	7.35	135·5
5.40	130·1	6.40	136·4	7.40	132·8
5.45	125·6	6.45	137·7	7.45	129·2
5.50	123·8	6.50	138·2	7.50	127·4
5.55	123·8	6.55	137·7	7.55	129·2
6.0	140·0	7.0	140·0	8.0	129·2

Total Revolutions run, 45,602. Mean Speed, 182·44.

REMARKS.—Magnolia bearing in perfect condition at finish; weight same as at commencement.

NOTE.—In explanation of the decrease in temperature of magnolia metal during the last portion of the run it should be stated that this bearing was cast on a mandril and put in place rough, so that it had comparatively little bearing-surface at first, and when examined at the end of the test was found not to have a full bearing even then; but, of course, as the total pressure remained the same throughout the test, viz. 3,024 lbs., as the bearing-surface increased the pressure per square inch was correspondingly reduced, hence the decrease in temperature.

With regard to the conditions under which this trial was made, we would point out that the pressure per square inch was not excessive, 600 lbs. being often used in Admiralty work; again, the velocity was also an average working allowance. But the remarkable point was the unusually long time the bearing ran without any supply of oil. Once more we may say we consider it a very great advantage, and worthy of special consideration, that under very fair working conditions a run was made lasting 4 hrs. 10 mins., during 4 % of which time only oil was supplied.

The last test to which we draw our readers' notice is, however, far more severe than either of the two foregoing. A number, thirty or forty, of different anti-friction metals were tested. The machine used was a 5-in. collar keyed on a 3-in. shaft, and lubricated with sperm oil. Speed was 1,600 revolutions per minute, giving a lineal velocity at surface of journal, 2,083 feet per minute, for all the metals.

Magnolia	ran	10	mins. with	1,000 lbs.	per sq. in.
”	”	20	”	1,200 lbs.	”
”	”	10	”	1,400 lbs.	”
”	”	10	”	1,600 lbs.	”
”	”	10	”	1,800 lbs.	”
”	”	20	”	2,000 lbs.	”

Its condition at the finish was practically perfect. The next best was Stone's Navy White Bronze, which fused after running 14 minutes with 1,000 lbs. per square inch; Parson's White Brass No. 2 fused in 10 minutes with the same pressure, and Dewrance's "Babbitt" fused in 10 minutes with but 900 lbs. pressure. This test gives a factor of $\frac{2,000}{600} = 3\frac{1}{3}$ over the greatest permissible pressure in steam engineering.

CHAPTER X

BALL BEARINGS

Introductory—Size—Material—Test of Balls—Design of Ball Bearings.

Introductory.—It is only during the past few years that the question of ball bearings has come into prominence. Cycles have been the pioneers in this movement in a manner somewhat similar to the way in which torpedo boats have taken the lead in high speeds, and become the pioneers in the production of those magnificent vessels which we justly term "ocean greyhounds." Conservatism in the methods of machine design, without doubt, has retarded their usage in this country. Well would it be for us if this stopped at ball bearings; but, unfortunately, it extends itself to other classes of machinery and tools.

A practical example of the importance of ball bearings in reducing friction will not be out of place.

Mr. W. R. Warner, of the noted firm of Warner and Swasey, U.S.A., at a topical discussion of the American Society of Mechanical Engineers, at Philadelphia, November, 1887, referred to the work of his firm in connection with the Lick telescope. The tube weighed 4½ tons; it was 56 ft. long. As is known in such constructions, the tube must be supported from one side—that is, it cannot be held by trunnions, but it must be supported near the centre and from one side.

It is necessary for this tube to work very easily; in fact, the friction, as far as possible, must be removed, as the tube must move very smoothly in following the star at which the observer is looking. It seemed almost impossible to make it move easily enough in the ordinary way by using friction rolls, so instead of that we used (said Mr. Warner) the method of surrounding the axis close to the tube with a series of rolls 2½ in. diameter and 3 in. long, with a result which seemed very satisfactory. This tube, weighing 4½ tons, when balanced on these rolls, would turn by a pressure of four pounds at the end (one finger would move it very easily), so that the problem was thus completely solved.

Another effort to solve a similar problem in a different position where the rollers would hardly do—for it was the end thrust on a heavy weight—was accomplished by using hardened steel balls running in circular concave tracks. In this problem, simply to test its working, Mr. Warner added, "We placed a weight of 2½ tons on forty balls 1 in. diameter in the two circular tracks, and this 2½ tons was turned by a pressure of one pound at a radius of 3 ft. Almost all the friction in these two cases was reduced to rolling friction instead of sliding friction."

Mr. Warner added that the balls followed the ring in which they were running. The groove in which they ran was $\frac{1}{2}+\frac{1}{32}$ in. radius, so that it was practically a plane surface, bearing only on the top and lower edge, and the balls worked together so that the whole ring, when they were pressed together, left only $\frac{1}{32}$ in. between the last two balls. In the case of the rolls, they were not together, but had their axis running on steel balls four-tenths of an inch diameter, the 2½ in. rolls thus being held in a live ring, so that their surfaces did not touch each other. There was no lubricant, everything ran dry. The best result was with the balls. As to the amount of weight allowed, it was found safe to put on the balls something less than 1,000 lbs. per ball, while on a roll having its bearing surface its full length of 3 in., a much larger weight could be laid.

Again, with regard to actual tests, Mr. G. F. Simonds, who has spent much thought and ingenuity upon the production of perfect balls and the design of ball bearings, said that some bearings of his have been subjected to an exhaustive test, to determine their wearing qualities and saving in power. This test demonstrated that the friction is less than one-thirtieth of that developed by the best class of ordinary bearings now in use. The rotating shaft of the testing machine was 2⅛ in. diameter, and the balls used in the bearings were three-eighths of an inch diameter. The first series of tests demonstrated that the maximum speed attainable with the best ordinary bearings under 200 lbs. pressure was 100 revolutions per minute, owing to excessive heating of the journals and boxes. A ball bearing mounted on the same shaft at 1,000 revolutions, under a pressure of 2,800 lbs., did not heat at all.

In the second series of tests ball bearings were introduced into the machine, where the ordinary bearing in the first test had been used. The velocity of the shaft was then increased to 2,600 revolutions per minute, and it was demonstrated that increase of pressure on the bearings did not perceptibly increase the resistance (or friction), and with an increase in the velocity the resistance was so slight as to be of no material importance.

In considering the subject, which we have introduced by the above interesting references, let us first look at two important requisites in good balls: (1) Size, and (2) Material.

Size.—In addition to the relation of one ball to another there is a most important feature to be attained first in each ball, apart altogether from its relation to any other—each must be a perfect sphere, or it will not roll equally well in every direction. It is owing to the difficulty found in turning perfect spheres in workshops which are not equipped with plant designed for that particular service, that at one time rollers used to find a more prominent place than they now do, for it was easy to turn a perfectly true cylindrical or conical roller, or a series of them, differing but an infinitesimal amount from each other with ordinary lathes. But the demand for balls has increased so much of late that it has justified the installation of special plant for the purpose of producing them, and consequently rollers do not find so much place. The next consideration with regard to size is that all the balls must be of exactly the same diameter if intended to work together, since an appreciable difference in diameter of any one of them will throw all the load on to the largest ball or balls.

The permissible amount of variation in the diameters of balls intended to work together is not by any means fixed on a uniformly accepted standard. One English firm guarantees, for example, no greater variation than $\frac{1}{2000}$th of an inch, while an American manufacturer claims for his productions a maximum of only $\frac{1}{10,000}$th of an inch. We have no doubt whatever that the first firm could produce balls equal in uniformity to those of the American firm, but of course it means more labour and care in manufacture, and consequently a higher selling price. We are sorry we cannot compare the prices of these two firms. This would be most interesting.

Balls are now made and sold for mechanical purposes of all diameters, from $\frac{1}{16}$ of an inch up to 2 ins. The few who have had experience with the design of ball bearings for other purposes than cycles, recommend as large a diameter as possible. The necessity for keeping everything light has necessitated the use of small balls on cycles, but where lightness of construction is not such an essential condition, the precedent of cycle manufacturers can be safely departed from.

A correspondent[1] in the *American Machinist* writes:—"I would always use balls of as large a size as reasonably practicable. The small sizes used for bicycles no doubt assist in keeping down dimensions, and therefore weights, but for ordinary machinery I think larger diameters are preferable."

Material.—For the work to which balls are put the material must, of course, be suitable, and must not only be hard and capable of resisting wear and tear, but also possess sufficient elasticity to return to their original shape after having been compressed by the bearing weight coming upon them. These conditions are fulfilled best by steel balls suitably hardened. Some makers use the very best close-grained crucible or diamond tool steel, heated in an American gas furnace, and hardened right out in water, and drawn down afterwards to the required degree of hardness. Other makers use Bessemer or low carbon steel, which costs one-fourth the price, and case-harden the balls—a method which, when applied to armour plates for warships, is known as the Harvey process.

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[1] Mr. W. H. Booth.

Tests of Balls.—The next point after having obtained a perfectly-shaped ball and of suitable material is to know what weight certain size balls will support and work well under. As may be imagined, very few reliable tests have been carried out, and fewer still have been made public. Mr. G. F. Simonds, who has patented and perfected several important inventions relating to the manufacture of balls, gave the following interesting facts at a meeting of the Franklin Institute, Mass., U.S.A., October, 1893: "A test of the resisting power of steel balls was made. Three $\frac{3}{8}$ in. diameter steel balls stood a direct pressure of 175,000 lbs., or 58,000 lbs. per ball, without injury. The same size balls were rotated between parallel plane surfaces, and under these conditions the balls and the bearing surfaces, the latter of Bessemer steel case-hardened, stood successfully a pressure of 2,500 lbs. per ball." The first figure (58,000 lbs. per ball) seems a little startling, and we cannot imagine such an extreme test leaving the balls exactly as it found them. We do not say that they were cracked or crushed, but certainly they must have had a permanent set in them which could have been detected with a micrometer gauge. We base this opinion upon some actual experiments made upon hardened steel roller surfaces which are lying before us, and which, though interesting, we have not space to reproduce.

Mr. Booth, whom we have already quoted, says: "For smaller bearings I have used $\frac{5}{16}$ in. and $\frac{3}{8}$ in.

balls, and have applied in this way about twenty balls of $\frac{5}{16}$ in. diameter to the repeated stress involved in lifting a load of some 5,000 lbs. or more in rock-boring operations."

Again, as stated above, Mr. Warner carried a load of 2½ tons on 40 one-inch diameter balls—that is, about 140 lbs. on each ball; but it was found that something less than 1,000 lbs. could be placed on each with safety.

As a general principle, small balls should be used for high speeds and light loads; and the heavier the load, irrespective of speed, the larger the balls must be. A ¼ in. diameter ball is found to crush at 6,700 lbs., and about one-tenth to one-fifteenth of this pressure may be taken as a safe working load, though it is always advisable, especially in horizontal bearings, to allow a fair margin, even over this figure. These proportions apply to slow-moving bearings; where the revolutions are above 30 or 40 per minute the load per ball should be decreased. For working figures the reader is referred to Table XX. The larger the balls used, the greater should be the margin, as the risk of internal stresses being set up in hardening is very greatly increased. In the smaller sizes—up to about ¾ in. diameter—this trouble is not found to any extent. The approximate crushing weight of balls varies as the square of the diameter:—

Diam. of ball. ins.	Crushing load in lbs. lbs.
¼	6,700
$\frac{5}{16}$	9,000
⅜	14,300
½	26,500
⅝	48,500
¾	67,000
1	92,500
1½	165,000

Design of Ball Bearings.—Having glanced at the question of the balls themselves, we can now proceed to the consideration of the best manner of applying them for reducing friction in mechanical bearings. Many people suppose that *sliding* friction is entirely eliminated in the use of ball bearings, and that rolling friction alone remains. But this is not the case, for not only does sliding friction take place through the balls touching and sliding past one another, but a certain amount also occurs between the balls and the surface they run upon. This may be very excessive in badly-designed work, but in good designs may be nearly eliminated.

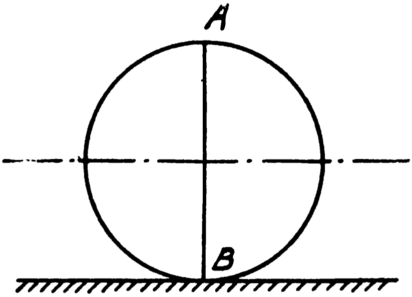

FIG. 45.

If a perfect sphere rests on a plane surface (see Fig. 45), the place of contact will mathematically be a point A, and if it rolls it will do so upon a line A, B. Now suppose a considerable weight were placed upon it so that yielding takes place due to the elasticity of the material used. Then the ball will run in a groove as shown in Fig. 46 (for the sake of argument we will suppose the bearing surface only to yield). If the ball is 1¼ in. diameter, and it turns round once about the horizontal axis A, B, the ball will have advanced $1\frac{1}{4} \times 3{\cdot}1416 = 3{\cdot}927$ in.—that is, the length of the circumference at C, D. The lines E, F, G, H at the outside edges of the groove are 0·8 in. long, and the circumference there is consequently $0{\cdot}8 \times 3{\cdot}1416 = 2{\cdot}513$ in., so that slipping to the amount of the difference of these two lengths will take place. In all probability a mean of the two lengths would be the amount the ball will travel in one turn.

In some early designs, the form of the grooves for the balls to run in was such that they fitted the balls as shown in Fig. 46. This was incorrect, as the above reasoning shows plainly. The balls should either run upon plane surfaces or in V-shaped grooves, where the ball will be in contact with the races at two points only, as at A and B (Fig. 47); a convenient angle for the groove is 45 deg.

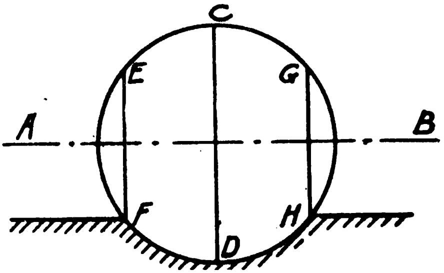

FIG. 46.

If the bearing is to be one to sustain end pressure—commonly called a thrust bearing—instead of a plain bearing for a horizontal shaft, then the form of groove shown in Fig. 47 will require to be altered somewhat, since the path of the points A, A in revolving round a point to the left of the ball as a centre will be shorter than the path in which the points B B will move. A good method for correctly setting out the form of a groove to overcome this difficulty is as follows:—Take any straight line, A, B (Fig. 48), and consider A as the centre of the circular race required; mark off A, G equal to the radius of centre of race, and draw C, D perpendicular to A, B. With G as centre, draw two circles; one equal in diameter to the size of ball to be used, and the other with a diameter three-fourths of the first; from A draw two lines tangential to the inside circle. These lines will cut the outer circle in four points, at each of which draw a tangent. These four tangents will form a four-sided figure in which the ball will roll without slipping, since these four points lie upon a cone whose apex is at A. The proportion between the length of the lines A, H and A, J is the same as that between H, K and J, L, and we have the same conditions, so far as relative velocities are con-

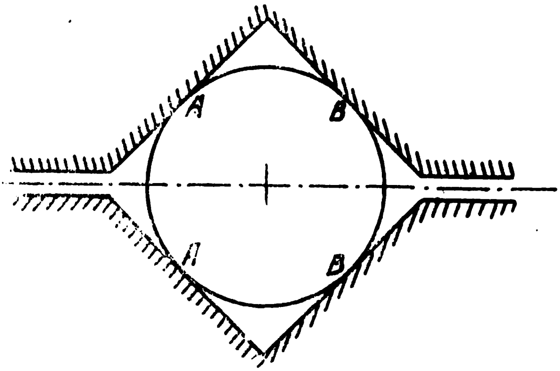

FIG. 47.

cerned, as if we had a conical roller rolling between the surfaces represented by the lines A, J, A, L, and it is obvious that rolling friction only would be present,

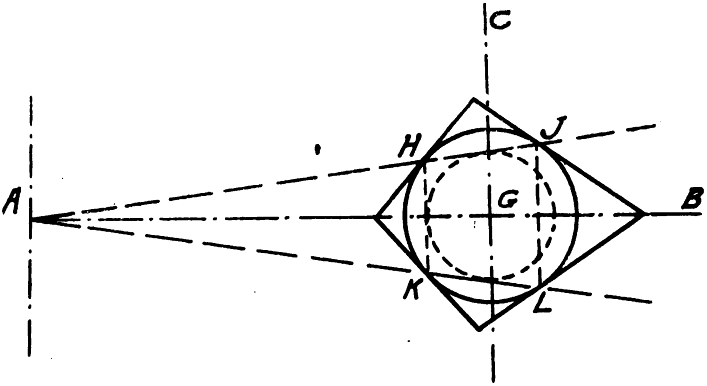

Fig. 48.

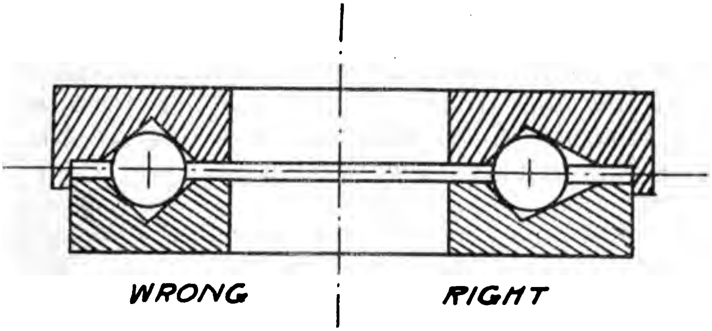

Fig. 49.

sliding friction being entirely eliminated. So far as we can learn, the necessity of this construction was first pointed out by Mr. Fred. J. Miller in the issue of the *American Machinist* of April 3, 1890.

In Fig. 49 we have shown a thrust bearing similar to those used for drilling machine spindles; the left-hand half of the diagram shows how the groove is often (and wrongly) designed; the other half how it should be designed. The material of the race plates should be similar to that from which the balls are made and hardened in the same way, care being taken that there are no flaws or soft places, for the reason that such will soon become holes, rendering the bearing worse than useless, for unless a ball bearing is very carefully designed and manufactured, it is far worse than an ordinary one.

For *heavily-loaded horizontal* shafts, ball bearings are not well suited, since all the weight has to be borne by a few—at the most one-fourth—of the balls at the bottom of the bearing, and the passing of the balls past the lowest point tends to a jerky motion. It will have been noticed that in thrust bearings, whether vertical or horizontal, each ball gets its fair share of the work. Such a combination of rollers and balls as was illustrated in the issue of *Engineering* of April 14, 1893, the invention of Messrs. Purdon and Walters, seems to be better suited to *heavy horizontal* shafts.

CHAPTER XI

BALL BEARINGS—ACTUAL PRACTICE

HAVING directed attention to the theoretical considerations involved in the design of ball bearings, we now bring before our readers various devices carried out in practice by manufacturers.

We will therefore revert to the work of Mr. G. F. Simonds. This gentleman perfected a metal-rolling machine some years ago, which is manufactured by The Simonds Rolling Machine Co., U.S.A. While at the Paris Exposition he had his attention directed to the subject of ball bearings and the ease with which his metal-rolling machine could be made to manufacture the balls. In the end of 1893 Mr. Simonds read a paper before the Franklin Institute, Mass., U.S.A., describing the long and painstaking methods by which he had arrived at satisfactory solutions of the problems which had presented themselves to him while conducting his experiments. In the course of his remarks he said that steel balls are now being made of absolute sphericity, like one another, and with great economy, and a system has been developed for making tempered articles uniform in quality, with a great saving in cost over existing methods for doing similar work. The accuracy and efficiency of the new system of making spheres will be appreciated when it is stated that balls of $\frac{1}{8}$th, $\frac{1}{32}$nd, and $\frac{1}{64}$th of an inch in diameter have been ground from pieces of steel, cylindrical in form, so that they do not vary $\frac{1}{10,000}$th part of an inch from one another, or from perfect spheres. The new system of making the tempered sleeves and rings for the balls to run in, and also for tempering the balls, is full of interest, but we must leave it, remarking only that a low carbon steel—Bessemer, for instance—can be used in the place of the high grade steel that is required when ordinary methods of tempering are adopted. By means of a special furnace the ancient and well-known cementation case-hardening process is brought under complete control. This is very important, especially as it enables the sleeves and rings in which the balls run to be case-hardened, or carburized to any given required depth and to any high degree of carbon, whilst it ensures that any number of such pieces will be of the same degree of hardness. We are given to understand that when manufacturers employ high carbon steel, and harden in the ordinary way, the risk in hardening large size balls, 3 inches or so, is so great that hardening is dispensed with; steel of extreme natural hardness being used instead, from which to make the balls, and left in its natural condition to withstand the wear. We give as briefly as possible some points in reference to the design of bearings for bicycles, cars, carriages, etc., which are worthy of a place in this chapter.

(1) In the ordinary design of bearing for this class of work the balls are loose, and must be handled singly.

(2) Each ball is required to sustain both the weight and end-thrust.

(3) Only two rows of balls can be used, one row at each end of the bearing.

(4) On account of the principle on which it is constructed the balls and bearing surfaces wear away.

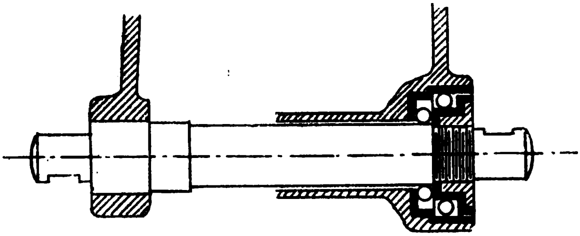

FIG. 50.

(5) There must be a means of adjustment on account of this wear.

In Mr. Simonds' design (see Fig. 50), what he calls the "double-case bearing," these conditions are reversed.

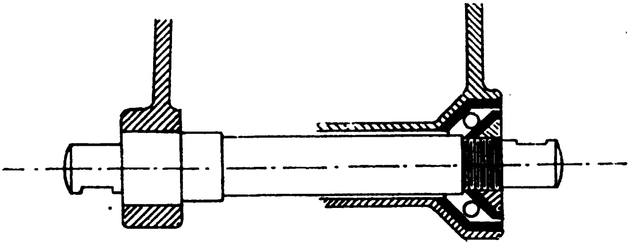

FIG. 51.

The balls are confined in cages, and readily and safely handled; one set of balls carries the weight, while another set takes the end-thrust; any required number of balls can be utilized; properly made, there is practically no wear, and as there is no wear of the parts, no adjustment is necessary. In Fig. 51 we show Mr

Simonds' "bevel-cage bearing," in which the balls run between plane parallel surfaces in place of grooved surfaces, although it is open to the objection that each ball carries both weight and end-thrust. It is intended for very light work, whereas the double-cage bearing is for use, where weight is not so much looked at. In our reproductions of Mr. Simonds' illustrations we have not shown the cages; suffice it to say that they are a light framework which does not hamper the balls.

We now turn to an English firm[1] who have gone in for ball production on a large scale, and who have laid down special plant for this particular purpose at Coventry. The machines used for the production of the spheres are automatic in their action, steel wire of a suitable diameter being fed through a hollow mandrel; and by means of a series of cutters which successively operate upon the wire, the balls leave the machine within 0·003 inch of their finished sizes, having been shaped and separated from one another. The next process is lapping, which is carried out in another department—emery and a mineral oil are used for this purpose. The spheres on leaving this stage are within 0·0005 inch of perfect accuracy. After the lapping comes the hardening process, and then they are again ground and finally polished with rouge. At various stages of the manufacture, notably at the end, the sizes of the balls are tested by automatic testing machines. These machines consist of steel bars placed at certain distances apart, down which the balls roll. By placing these bars at distances apart, varying half of a thousandth part of an inch in consecutive machines, the balls are sorted out into parcels, the maximum variation of any one ball in any parcel being 0·0005 ($\frac{1}{2000}$th) inch. After this a microscopic examination is made of every ball in order to find any flaws that may exist, and which are at once revealed by means of the polishing. We are given to understand that 80,000 balls are turned out from these works per diem. Where they go may be a cause of wonder to many, but it clearly shows the growing use of balls when they can be thoroughly relied upon.

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[1] The Auto Machinery Co., Ltd., Coventry.

In the production of these balls the finest crucible cast steel—diamond steel—is used, costing between 9*d.* and 10*d.* per lb. Great care is of course necessary and used in the process of hardening. The balls are thrown into a large tank of water after being heated in a special furnace, so that all may be of the same temperature before being cooled out.

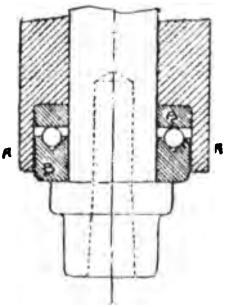

FIG. 52.

In Fig. 52 we show a section of a bearing intended to take the end-thrust of a drilling machine spindle. One feature is worthy of attention, the way in which the bearing is recessed into the spindle, the outside flange A, A serving to keep it dust proof. It will be noticed, too, that the top and bottom races are separate pieces of hardened steel in keeping with the best practice.

Another bearing designed for a horizontal shaft is shown in Fig. 53; this has only one set of balls, but the number of rows may be increased when it is necessary.

As examples of high speeds and light loads we may mention a wood-carving machine spindle in one of the largest ship-yards of Scotland, running continuously at a speed of 13,000 revolutions per minute, perfectly cool and with little or no oil, the plain bearings originally fitted having given constant trouble. Again, a small electric motor, which originally ran very unsteadily at 6,000 revolutions per minute, when fitted with balls ran at 7,000 with the same amount of current, and so steadily that all sparking at the brushes was

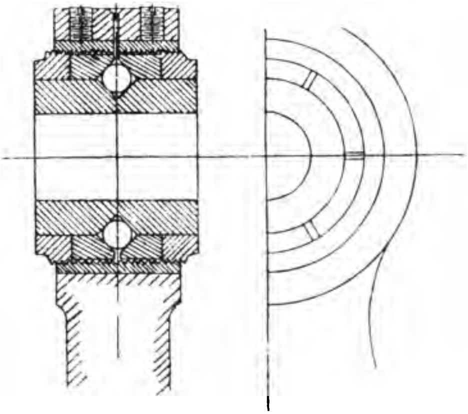

FIG. 53.

done away with, and it was found possible to grind a razor perfectly on an emery wheel attached to the spindle.

In July of 1894, at the Manchester meeting of the Institution of Mechanical Engineers, the late Dr. John Hopkinson read a paper descriptive of the New Electric Lighting Works at Manchester, and during the discussion following the paper it was mentioned that ball bearings, supplied by the Auto Machinery Company of Coventry, had been fitted to the motor generators with the view both of economizing power by reducing friction, and also of overcoming, as far as possible, any liability to stoppage on the part of the workmen in charge of the machines. A small dynamo fitted with these bearings had been running at 1,300 revolutions per minute continuously for nine hours, without oil or any attention whatever, and showed no signs of heating. Readings taken from the ammeter every 15 minutes during the running showed a gain in pressure of about 10 per cent. over similar readings taken the day before, under as nearly as possible the same conditions, when the dynamo was fitted with ordinary bearings. It is of interest to note the ready means of adjustment, and also the ease with which they can be taken to pieces and put together again. The adjustment is provided for by the nuts A, A (see Fig. 54), which lock each other in the same way as they do on an ordinary lathe mandrel, and are tightened until the sleeve B, B turns freely without the slightest shake. In accordance with the principles of design, which we have before spoken of, the balls do not run directly on to the shaft C, C, but on the hardened steel sleeve B, B, which is caused to rotate with the shaft by means of the key D. This also enables the balls to be placed in position in the grooves before they are put on the shaft, which saves a great amount of annoyance and inconvenience.

Mr. Renouf, of Birmingham, instanced, in the course of a discussion in *Engineering*, at the end of 1893, various cases of balls being used for thrust bearings, in reply to some adverse criticisms. At Humber and Co.'s Beeston works a heavy milling machine has now been running for ten years with ball thrusts. At the Coventry works of the same firm two vertical pulleys, carrying a belt weighing 2 cwt., and transmitting 40 horse-power, were a constant source of trouble till a thrust bearing, with $\frac{3}{4}$ in. balls, was fitted. They are now working well. This gentleman, by the way, also considers that it is unadvisable to have fewer than nine or more than twenty balls in a row, best results being obtainable with from eleven to fifteen.

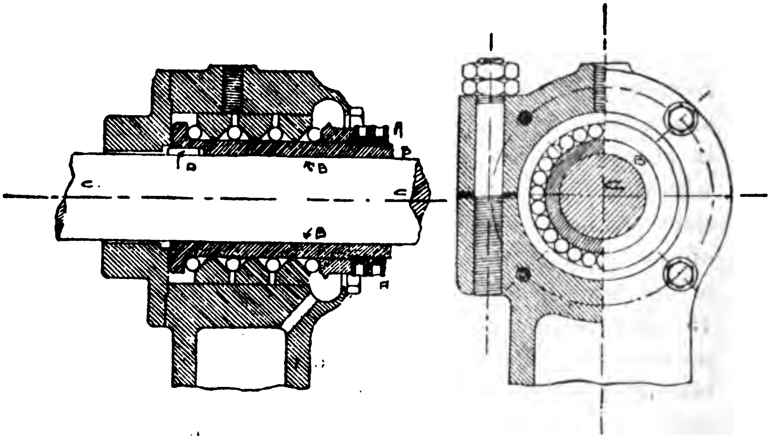

FIG. 54.

TABLE XX.

Price each Bearing.	No. of Bearing.	Diam. of Shaft.	Extreme diam.	Extreme Thickness.	Diam. of Balls.	No. of Balls.	REVOLUTIONS PER MINUTE.													Load per Ball.	Extreme safe load per ball without permanent set.	Crushing Load per Ball.
							10	50	100	150	200	300	400	500	600	700	800	900	1000			
							SAFE WORKING LOAD IN LBS.															
		Ins.	Ins.	Ins.	Ins.																Tons.	Tons.
17/6	1	$\frac{3}{4}$	$1\frac{3}{4}$	$\frac{5}{8}$	$\frac{5}{16}$	11	2142	960	680	554	490	392	340	300	277	256	240	226	214	618	...	...
20/-	2	1	$2\frac{1}{4}$	$\frac{3}{4}$	$\frac{3}{8}$	13	3120	1400	985	805	697	570	494	442	402	378	349	329	312	757	$1\frac{1}{2}$	6
22/-	3	$1\frac{1}{4}$	$2\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{8}$	15	3350	1580	1060	865	750	612	530	475	434	400	375	354	335	707	...	...
24/6	4	$1\frac{1}{2}$	$2\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{16}$	16	5850	2620	1850	1470	1310	1070	925	830	756	700	655	618	585	1156	2	8
29/-	5	$1\frac{3}{4}$	$3\frac{1}{8}$	1	$\frac{1}{2}$	16	7460	3320	2360	1930	1670	1360	1180	1055	963	892	834	786	746	1475	3	12
33/-	6	2	$3\frac{3}{8}$	1	$\frac{1}{2}$	17	7580	3400	2400	1960	1700	1390	1200	1070	980	906	850	800	758	1411	...	...
36/6	7	$2\frac{1}{4}$	$3\frac{3}{4}$	$1\frac{1}{8}$	$\frac{9}{16}$	16	10400	4650	3290	2682	2325	1900	1645	1470	1345	1245	1160	1100	1040	2056	4	16
39/6	8	$2\frac{1}{2}$	$4\frac{1}{4}$	$1\frac{1}{4}$	$\frac{5}{8}$	17	16000	7150	5160	4130	3570	2920	2530	2250	2065	1910	1785	1720	1600	3034	5	20
44/-	9	$2\frac{3}{4}$	$4\frac{1}{2}$	$1\frac{1}{4}$	$\frac{5}{8}$	18	16600	7540	5260	4300	3720	3030	2630	2350	2140	1985	1855	1750	1660	2922	...	...
51/-	10	3	5	$1\frac{1}{2}$	$\frac{3}{4}$	17	25700	11500	8140	6550	5750	4700	4060	3640	3320	3070	2880	2710	2570	4790	$7\frac{1}{2}$	30
57/-	11	$3\frac{1}{4}$	$5\frac{3}{8}$	$1\frac{1}{2}$	$\frac{3}{4}$	18	26400	11800	8350	6800	5900	4830	4170	3740	3410	3160	2960	2780	2640	4640	...	...
66/-	12	$3\frac{1}{2}$	6	$1\frac{3}{4}$	$\frac{7}{8}$	17	37100	16600	11720	9500	8250	6760	5850	5250	4790	4430	4140	3910	3710	6894	9	35
71/-	13	$3\frac{3}{4}$	$6\frac{3}{8}$	$1\frac{3}{4}$	$\frac{7}{8}$	18	39282	17576	12410	10058	8725	7158	6194	5859	5072	4690	4384	4140	3925	6894	...	...
80/-	14	4	7	2	1	17	46500	24000	14700	11560	9800	7500	6570	6210	5360	4975	4660	4400	4160	8048	$10\frac{1}{2}$	41

The above illustrations and Table XX. show the safe working loads for horizontal and vertical thrust bearings. They give, too, the safe working loads at given revolutions per minute, arrived at from actual tests made at the manufacturers' works. They are intended for use in connection with worm gearing, crane posts, turntables, crane hooks, turbine, propeller and other shafts.

Fig. 55. Fig. 56.

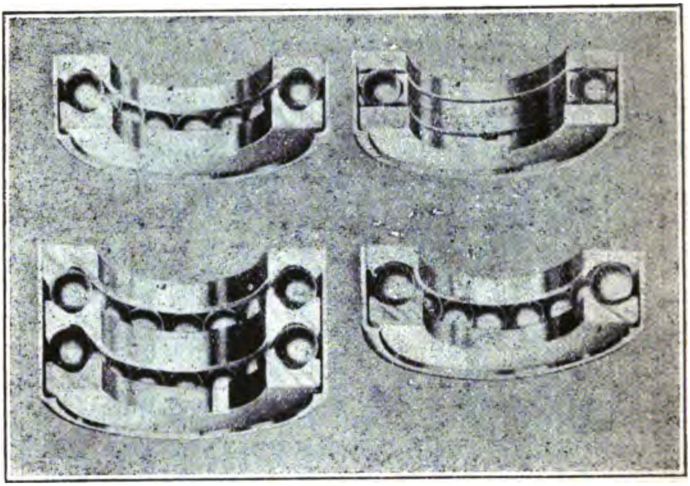

Fig. 57. Fig. 58.

The figures in Table XX. apply to Fig. 58; but for high speeds Fig. 57 is recommended in preference, as the two sets of balls halve the speed at which they travel relatively to the races in which they move. Figs. 55 and 56 are for use with vertical shafts. Fig. 55 will carry 75 per cent. and Fig. 56 will carry 50 per cent. of the loads given in the Table for Fig. 58.

The prices given above are for Fig. 58, to which should be added one-third or $33\frac{1}{3}$ per cent. for Fig. 57, while 10 per cent. is to be deducted for Fig. 55, and 15 per cent. for Fig. 56. In all cases the dust caps are extra, the races being of different forms—Figs. 57 and 58 being designed as shown on opposite page, and Figs. 55 and 56 being round grooves and flat plates respectively.

TABLE XXI.

No. of Bearing.	Diameter of Shaft. Inches.	External Diameter. Inches.	PRICES.	
			Ball Cage Bearing.	One Cast Steel Ring.
1	$\frac{1}{2}$	$1\frac{1}{8}$	3/6	1/6
2	$\frac{5}{8}$	$1\frac{1}{4}$	3/8	1/7
3	$\frac{3}{4}$	$1\frac{3}{8}$	3/10	1/8
4	$\frac{7}{8}$	$1\frac{5}{8}$	4/-	1/9
5	1	$1\frac{3}{4}$	4/2	1/10
6	$1\frac{1}{8}$	$1\frac{15}{16}$	4/4	1/11
7	$1\frac{1}{4}$	$2\frac{1}{16}$	4/6	2/-
8	$1\frac{3}{8}$	$2\frac{3}{16}$	4/8	2/1
9	$1\frac{1}{2}$	$2\frac{5}{16}$	4/10	2/2
10	$1\frac{5}{8}$	$2\frac{7}{16}$	5/-	2/3
11	$1\frac{3}{4}$	$2\frac{9}{16}$	5/2	2/4
12	$1\frac{7}{8}$	$2\frac{11}{16}$	5/4	2/5
13	2	$2\frac{15}{16}$	5/6	2/6
14	$2\frac{1}{4}$	$3\frac{1}{4}$	5/8	2/7
15	$2\frac{1}{2}$	$3\frac{1}{2}$	5/11	2/8
16	$2\frac{3}{4}$	$3\frac{3}{4}$	6/2	2/9
17	3	$4\frac{3}{16}$	6/5	2/11
18	$3\frac{1}{4}$	$4\frac{7}{16}$	6/8	3/1
19	$3\frac{1}{2}$	$4\frac{11}{16}$	6/11	3/3
20	$3\frac{3}{4}$	5	7/2	3/6
21	4	$5\frac{1}{4}$	7/6	3/9

A very simple form of thrust ring is illustrated in Fig. 59, and is designed for drilling machine spindles, lathe mandrels, motor-car transmission boxes, etc. They are designed for lighter loads than those illustrated in

Figs. 55 to 58. The ball cage is shown above, and consists of two steel rings, held together with small pins.

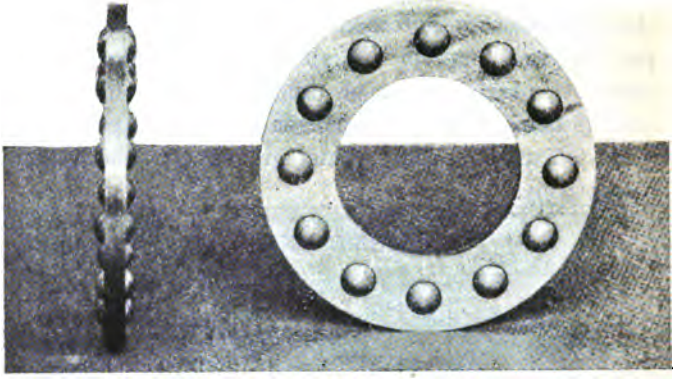

Fig. 59.

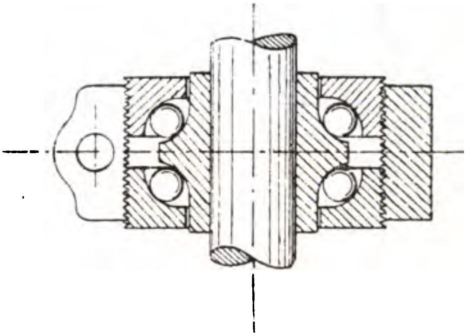

Fig. 60.

The use of balls for reducing friction in crane hooks is shown in Figs. 60 and 61. The races are shown circular in form ; they are often made especially for large sizes with tangential races, as in Fig. 49, which has also a dust lip on the upper race.

TABLE XXII.

No.	Price per set. £ s. d.	Load in tons.	A.	B.	C.	D.	Size of Balls.	No. of Balls.
1	1 7 6	1	$1\frac{1}{8}$	$1\frac{5}{32}$	$2\frac{1}{4}$	$1\frac{3}{8}$	$\frac{3}{8}''$	13
2	1 13 0	2	$1\frac{3}{8}$	$1\frac{13}{32}$	$2\frac{7}{8}$	$1\frac{13}{16}$	$\frac{1}{2}''$	13
3	1 18 6	3	$1\frac{5}{8}$	$1\frac{21}{32}$	$3\frac{1}{8}$	$1\frac{13}{16}$	$\frac{1}{2}''$	14
4	2 3 6	4	$1\frac{13}{16}$	$1\frac{7}{8}$	$3\frac{11}{16}$	$2\frac{1}{8}$	$\frac{5}{8}''$	13
5	2 7 6	5	2	$2\frac{1}{16}$	$3\frac{13}{16}$	$2\frac{1}{8}$	$\frac{5}{8}''$	14
6	2 12 0	6	$2\frac{1}{8}$	$2\frac{3}{16}$	4	$2\frac{1}{4}$	$\frac{5}{8}''$	15
7	2 17 0	8	$2\frac{1}{4}$	$2\frac{5}{16}$	$4\frac{1}{8}$	$2\frac{1}{4}$	$\frac{5}{8}''$	16
8	3 3 6	10	$2\frac{1}{2}$	$2\frac{9}{16}$	$4\frac{3}{4}$	$2\frac{7}{16}$	$\frac{3}{4}''$	14
9	3 10 0	12	$2\frac{3}{4}$	$2\frac{13}{16}$	$5\frac{1}{8}$	$2\frac{7}{16}$	$\frac{3}{4}''$	16
10	3 17 0	15	$3\frac{1}{4}$	$3\frac{5}{16}$	$5\frac{1}{2}$	$2\frac{9}{16}$	$\frac{3}{4}''$	18
11	4 19 0	20	$3\frac{1}{2}$	$3\frac{9}{16}$	$6\frac{1}{8}$	$3\frac{1}{8}$	$\frac{7}{8}''$	16
12	5 17 6	25	$3\frac{7}{8}$	$3\frac{15}{16}$	$7\frac{1}{8}$	$3\frac{1}{4}$	1″	17
13	6 17 6	30	$4\frac{3}{8}$	$4\frac{7}{16}$	$7\frac{1}{2}$	$3\frac{1}{2}$	1″	18
14	8 16 0	40	$4\frac{7}{8}$	$4\frac{15}{16}$	$8\frac{1}{4}$	$3\frac{5}{8}$	$1\frac{1}{8}''$	18

Table XXII. gives leading dimensions, sizes, and number of balls, so that the above can be readily incorporated with a design, and the prices give some idea of how much will be spent on securing the benefits of ball bearings—an item greatly needed by those in the drawing-office as well as by the customer in determining the advisability of their adoption under varying circumstances.

The combination of parallel and end-thrust bearings is shown well in Fig. 62, and is a type for use in tramcar, motor, lorry, and other work where their use is a great matter in the reduction of effort required for starting as well as when running. The writer leaves this chapter feeling how much more might be well written on the matter, but space forbids.

In conclusion, we again repeat that a poorly-designed

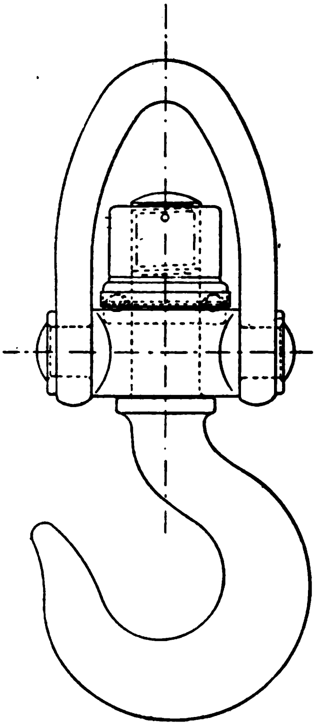

Fig. 61

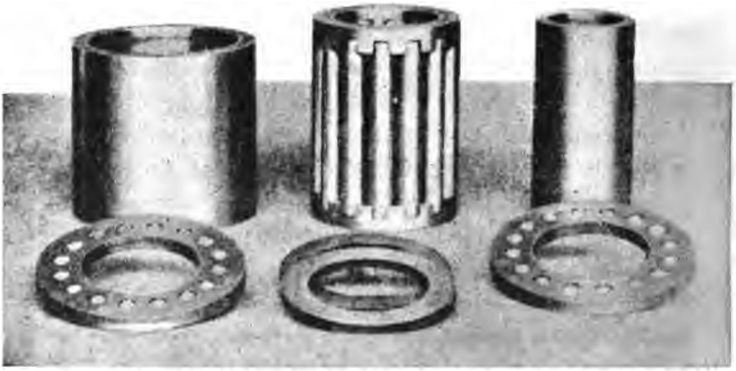

Fig. 62.

and badly-constructed ball bearing is far worse than the worst parallel bearing in existence, and that the objections urged so often against these friction-preventing devices seem to be founded upon some laxity in one or other, perhaps both, of these two pre-requisites.[1]

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[1] We gladly acknowledge our indebtedness to Messrs. the Auto Machinery Co., Ltd., Coventry, for the tests, etc., of steel balls; also for the use of their figures, Nos. 57, 58, 59, 60, and 61.

CHAPTER XII

FORCED LUBRICATION

At the Birmingham meeting of the Inst. of Mech. Engineers, July 1897, a paper[1] was read by A. Morcom, Esq., which is of great value in connection with our subject. It is a mechanical means for automatically flooding the engine bearings, the crank pins, the cross-head pins, and eccentrics with oil, and reproducing, as nearly as possible in actual practice, the conditions of the oil bath which gave such low co-efficients of friction as recorded on pages 29 and 30, due to the film of oil kept between the surfaces. Mr. Morcom's firm (Messrs. G. E. Belliss and Co.) came to the conclusion that a double-acting engine of ordinary construction, with a system of oil pump and pipes supplying oil under pressure to every bearing, would give (1) noiselessness in running at high speeds, (2) economy in maintenance owing to less wear and tear, and (3) a greatly increased B. H.-P. from the reduced engine friction.

These results, which had previously sought to be obtained by using single acting engines and running the connecting rod and other moving parts in a closed crank chamber partially filled with oil, were obtained with a marked degree of success.

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[1] High Speed Self-Lubricating Engines. *Proc. I.M.E.*, 1897.

The working parts, even when slack, worked without noise. The wear was imperceptible after long runs of months' duration. The mechanical efficiency, or

$$\frac{\text{B. H.-P.}}{\text{I. H.-P.}}$$

was as low as 96·3 under full load, and 89·3 under quarter load. These figures were so striking that Prof. Kennedy, who carried out some tests on a 200 H.-P. compound engine (12¼ in. and 20 in. × 9 in. running at 360 revs.) supplied to the Sunderland Corporation Electric Supply Station, said, "The most singular point about them is the extraordinary high brake efficiency of the engine, which is altogether much higher than anything I have previously found. I have been extremely suspicious of the figures, and have had everything checked repeatedly. The different figures taken at different periods are quite consistent with each other, and there is nothing in the records to suggest error, except the figures themselves. I can only place these on record without saying more about them."

When the reader remembers that for an ordinary engine (vertical) the efficiency at full load is not more than 92%, and at ¼ full load is often not more than 67% the value of this system of lubrication will be forcibly brought home to him.

The oil is pumped by means of a plunger attached by a pin to one of the main eccentrics, and the rocking movement is utilized to cause the pump body to move in a circular path over ports so that the valves of an ordinary pump are dispensed with. A pressure of about 15 lbs. to the square inch is maintained in the system.

The plunger is 1½ in. diameter, and the suction and delivery pipes are ¾ in. diameter. The use of large connections is most necessary as it helps to prevent clogging. It has been found best to introduce the oil at the top of the main bearings, and convey it by large channels to the sides and bottom. The oil is conveyed from them by means of long passages drilled in the crankshaft to the crank pin, crosshead pin, eccentric, and pin. It is returned to a reservoir in the bed of engine, and passes through an oil strainer on its way to the pump again.

The picture has been drawn of burst pipes, choked pumps, gagged valves, strainers that will not strain, etc., in connection with this system, but the writer's inquiries as to the behaviour of the engines designed by Mr. Morcom has been, on the contrary, quite favourable; but certainly the elements of danger are there, and it needs the best design and workmanship to prevent those untoward events happening.

The pump is placed at the bottom of the bedplate of the engine, just between the two inner bearings.

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